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HELICOPTER DRIVE SYSTEM R&M DESIGN GUIDE

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APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

Drive system components are among the largest contributors to Army helicopter reliability and maintainability problems. Past studies have focused on identifying the magnitude and nature of the problems. Other efforts have investigated the potential R&M benefits of specific design concepts. Extensive work has been performed in establishing the feasibility of "on-condition" maintenance. An extensive effort continues to address diagnostics. Much of the documentation of these endeavors has been written expressly for the R&M engineer--in a technical jargon incomprehensible to many designers.

The objective of this contract was to "translate" the aforementioned endeavors putting R&M into proper perspective, thereby making design engineers more conscious of the R&M aspects of the drive systems they design. The results are published in two reports: TR 78-50, Helicopter Drive System R&M Design Guide, and TR 78-51, a final report documenting the program.

The approach was to analyze failure modes experienced, contrasting them to current practices to determine design, development, and overhaul deficiencies. Analytical methods for estimating "off-the-board" reliability were reviewed. Testing methods, specifically accelerated testing versus overload testing, and reliability growth were addressed. On-condition maintenance and diagnostics were also addressed. Positions in two of these areas follow.

Reliability estimation needs more work to achieve parity with strength and weight analyses. Hazard functions could not be correlated with such design parameters as load or induced stress, precluding the assignment of "hard" numbers to reliability estimation. Probabilistic design is the best way to predict service life. It has the potential for optimum utilization of weight, and is the only means for setting realistic bounds on reliability problems involving costs, warranties, and producer's risk. The R&D necessary to bring probabilistic design "on stream" is encouraged.

Regarding diagnostics, fuzz burn-off chip detectors coupled with superfine filters are seen as the simplest, most cost-effective diagnostic system for modern helicopter drive systems. Fine filtration has the potential for rendering spectrometric oil analysis (SOAP) and particle count techniques obsolete. The Automatic, Inspection, Diagnostics, and Prognosis System (AIDAPS) requires very sophisticated instrumentation. Without a breakthrough in understanding the sympton-failure relationship, development of a practical, cost-effective AIDAPS system appears remote.

This program was conducted under the technical cognizance of Joseph H. McGarvey, Aeronautical Systems Division.

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This report is a reliability and maintainability design guide for helicopter drive systems. While not containing "how to" design information, the guide points out those areas of design which can be troublesome in the reliability and maintainability of helicopter drive systems. Besides containing information on the various drive system components, a management section is included that outlines some practices which design managers may employ to insure that reliability and maintainability are given proper emphasis during a design

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program. The final section is devoted to a step-by-step procedure for hazard function analysis, which may be used to predict the reliability of a gearbox during the design stage.

PREFACE

The design guide presented herein was prepared under Contract DAAJ02-76-C-0047 for the Applied Technology Laboratory, U. S. Army Research and Technology Laboratories (AVRADCOM), Fort Eustis, Virginia. Technical direction for the program was provided by Mr. J. McGarvey of the Applied Technology Laboratory. Mr. C. Keller served as Task Manager at Sikorsky. Technical contributors were Mr. H. Frint of the Transmission Design and Development Section and Mr. B. Trustee of the R&M Section.

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INTRODUCTION

In recent years there has been an ever-increasing emphasis on reliability and maintainability in the design of military hardware. The reasons for this increased emphasis follow:

1. Material and manpower costs are expected to maintain their upward spiral for the foreseeable future.

2. The branches of government overseeing military spending are becoming increasingly conscious of the costs of military equipment.

3. Maintenance is generally the single most important item in the life-cycle cost of military hardware.

The design guide presented here is part of the Army's effort to make drive system design engineers more conscious of the reliability and maintainability aspects of the gearboxes they design.

This design guide is essentially divided into three parts. The first part deals with planning and management of a drive system design effort. Although most engineers who head such design efforts are not trained managers, they are usually given little guidance in how to plan the design/ development program. In general they are given only drawing and test schedules, a budget requirement, and weight requirements. The tendency is to put too much emphasis on some of these program goals, usually at the expense of the structural reliability of the design. The first part therefore suggests various practices which may be implemented to avoid this pitfall.

The second part is the "design" part of the design guide. Unlike more conventional design guides, however, the information presented here is not basic "how to design" information. Rather, it is more concerned with the subtle aspects and details of design that often mean the difference between high and low reliability. Because the sections that make up the second part presuppose a basic knowledge of the design of helicopter drive system components, analytical equations and illustrations showing basic terminology will not be included. The emphasis of the second part is on "attention to detail", and as such will include design checklists which may be used to insure that all aspects of a design are given adequate attention.

The final part of this design guide presents the basic methodology for determining gearbox reliability with hazard function analysis. Although the numerical results obtained from such an analysis may not be correct in an absolute sense, the results can be correlated with a previous design whose reliability level is known. Knowledge of the relative reliability between the on-the-board design and the previous design can be extremely useful in insuring that the new design meets its reliability goal.

PLANNING AND MANAGEMENT OF THE DESIGN PROGRAM

The design of a reliable and maintainable drive system begins with planning and management that gives proper emphasis to both of these extremely important areas. If reliability and maintainability are not primary considerations of the design manager it is unlikely that the design he produces will achieve the desired reliability and maintainability goals. There are two factors, both present in any drive system design program, that could adversely affect the reliability and maintainability of the design: schedule and weight requirements. Every drive system manager is extremely conscious of these two factors because, first, they are extremely visible to upper management, and second, the design manager's performance rating may depend on how well he meets these requirements. Therefore, there is a great temptation for design managers to over-emphasize these areas, usually at the expense of other, perhaps more important, considerations, such as reliability and maintainability. This section contains a number of suggestions whose implementation can minimize the possibility of this happening.

PLANNING

Before serious design work begins, there is a substantial amount of planning with which the design manager is burdened. This planning usually takes the form of defining tasks, setting schedules and making man-hour estimates. However, this initial planning should also address whether technological developments such as rotor isolation will be an integral part of the design. If so, planning for it from the very beginning would enhance prospects for an optimum system. Rarely does anyone except the R&M engineer concern himself with reliability and maintainability at this stage; yet this is precisely where reliability and maintainability assurance should start. The following three tasks should be planned as an integral part of the design program to insure that reliability and maintainability do not become just incidental considerations:

1. Preliminary Reliability Analysis
2. Design Review
3. Detailed Reliability Analysis

Preliminary Reliability Analysis

After the basic drive system arrangement has been laid out and sized, but before detail design begins, a preliminary reliability analysis should be performed on the gearbox design according to the procedures outlined in the Hazard Function Analysis section of this guide. This analysis should then be compared to a similar analysis performed on an existing design with a known MTBF. In this way the reliability of the new design can be assessed very early in the design stage. If it appears from the analysis that the design will not achieve the desired goal, such design changes as switching to bearings with higher lives can be made with as little trouble as possible. Since this is a preliminary analysis, some simplifications will have to be made in the reliability analysis to keep the effort down to a reasonable level. The bearing portion of the analysis must be

simplified, since at this stage of the design such quantities as the lubrication, misalignment, and material factors have not as yet been determined. Hence, a uniform life adjustment factor, which can be estimated from past experience, should be applied to all of the basic bearing lives.

One additional item should be noted here with regard to the preliminary reliability analysis. The analysis should be performed based on the expected aircraft mission power spectrum. If, however, the drive system is to be subjected to a "must-pass" overstress test, it would be wise to also conduct the analysis using the test spectrum to determine what the probability is of passing the test. In recent years some of these tests have become so severe that designing the gearbox for a reasonable chance of passing the test is a more severe requirement than designing the gearbox for the required field reliability level.

Design Review

It is suggested that the design manager subject his basic design layout to the scrutiny of other drive system engineers before the start of the detail design phase of the program. He should give to other design engineers copies of the basic design layout with such information as shaft speeds and basic bearing lives, and should ask for their comments and criticisms of the design. The benefits of this type of design review cannot be underestimated. It is virtually certain that at least a few flaws in the basic design will be found by the reviewing engineerings. They will have had experience with one feature or another where past experience has shown that it should be changed. No single design engineering will have had experience with all of the past problems; hence, the more design engineers reviewing the basic layout, the smaller the chance that past mistakes will be repeated.

Detailed Reliability Analysis

This is essentially a repeat of the reliability analysis suggested earlier. This analysis, however, is performed when the program is far enough into the detail design stage to use actual bearing lives with the appropriate adjustment factors and with a refined power spectrum. Again, this analysis should be compared to a similar analysis on a gearbox with a known MTBF. It is also suggested that if a "must-pass" overstress test is required, the detailed reliability analysis also be performed using the test spectrum.

MANAGEMENT

Apart from incorporating the above suggestions as integral parts of the design program, there are a number of other rules that the design manager should abide by to ensure that reliability and maintainability goals are realized.

First, the design manager must keep in mind that reliability is the single most important cost consideration in drive system design. Maintenance and replenishment spares generally account for more than 50 percent of the life-cycle cost of an aircraft as shown by the life-cycle cost breakdown of Figure 1. Hence, even without considering the cost of aircraft downtime

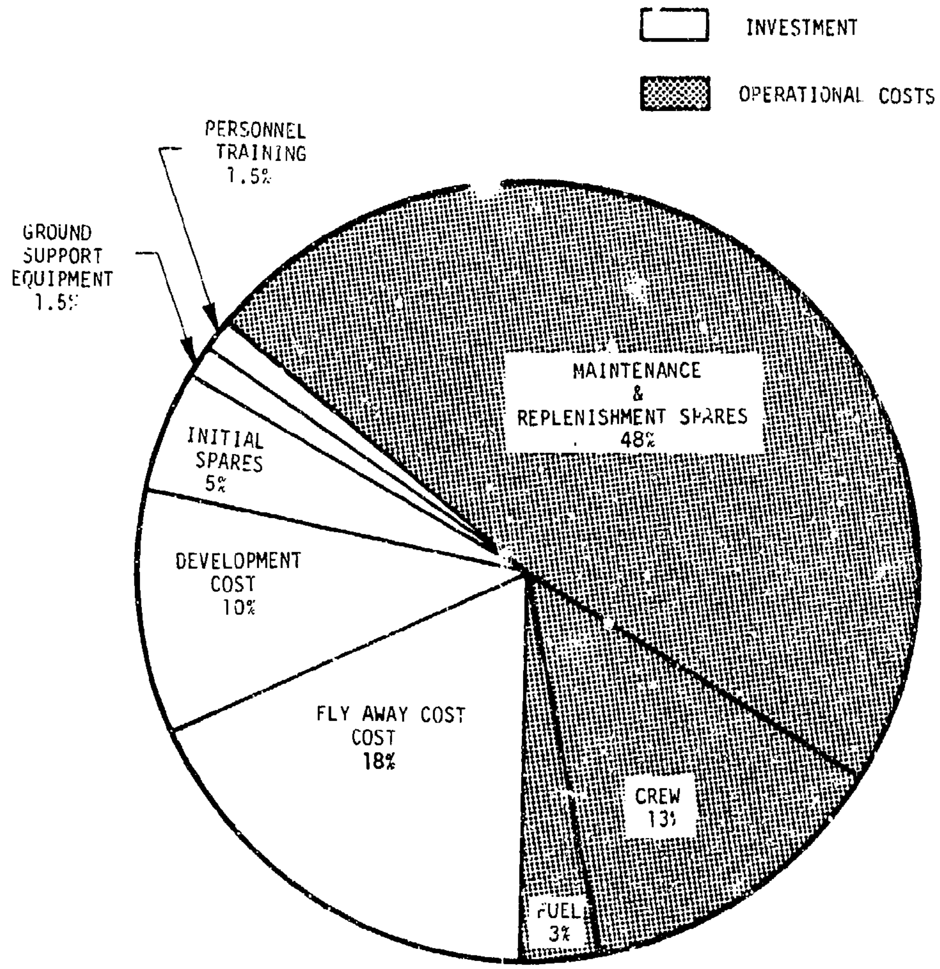


Figure 1. LIFE-CYCLE COST BREAKDOWN OF TYPICAL MILITARY HELICOPTER

or mission aborts, it is clear that the importance or reliability must not be underestimated.

With regard to executing the design program, the most important activity with regard to reliability is the analysis of the various drive system components. The analysis of a helicopter drive system should be very thorough with attention to detail given top priority. The analytical effort should not be sacrificed to maintaining schedule. No one ever remembers that the drawings went out on time or that the weight target was reached, when failures start occurring during development testing or field operation. One way to ensure that the analysis is thorough is to use design checklists such as those provided in this guide for the various drive train components. Although many of the items on these checklists seem obvious, it is surprising how often the obvious is overlooked. Reviews by other areas such as R&M, manufacturing, product support, and quality control can also head off many problems which may be more troublesome in later stages of the program. Changing lines on a drawing during the design stage is always easier and cheaper than redesigning existing hardware. It is also important that any high risk areas of a design be identified early in the design stage and that alternate designs be prepared in the event the initial design does not work out. This is especially true where new technology is being introduced for the first time into a production design.

Maintainability also requires much attention during the design stage. The most common problem with respect to maintainability in past designs has been a lack of accessibility to drive train components that required servicing. To minimize this, the transmission design manager should maintain close contact with the aircraft configuration manager and the design managers from other groups such as airframe, propulsion, and controls, etc., to make certain that hardware integral to those areas does not interfere with drivetrain maintenance and vice versa.

GENERAL DESIGN CONSIDERATIONS

Although the detail design of helicopter drive train components is perhaps more important in insuring the reliability and maintainability of a given drive system, some general design constraints and requirements should be given special attention because of their potential impact on the long-term reliability of the drive system. The first of these is the mission power spectrum. The key point with regard to this design parameter is to anticipate increases in power and to leave room for growth in all primary power train gears and bearings. This is especially important for tail rotor drive system components, where the design power may be much less than maximum continuous. Despite predictions by aerodynamicists, the power requirement for the tail rotor will not be known with certainty until flight testing begins. If no allowance is made for greater than predicted power levels, the redesign will be less than ideal, and the reliability of the components is likely to be compromised. Another point to remember is that virtually every military aircraft is uprated in power before the end of its operational life. If this is not anticipated during initial design, the reliability of the later uprated transmission will suffer. In other words, one should anticipate changes for the worse.

New developments in technology or the addition of new features must also be anticipated during initial design. For example, it is possible that a rotor isolation system may be added in the future. In this case the transmission designer must be certain that, if required, a new coupling with a greater misalignment capability can be added without drastically altering the present design. Allowance should also be made for increases in housing stiffness without interfering with the control system or the internal gearbox components. Room for growth in the filter envelope is another very desirable feature. The superfine filters currently under development will in all likelihood require larger envelopes to accommodate the increase in particles trapped in the element.

Another factor to consider is the detectability of failure modes within the gearbox. Care must be taken to insure that debris resulting from any possible failure can reach the chip detectors. A failure mode and effects analysis can be helpful in determining this. It will also point out where simple design changes can prevent relatively minor failures from becoming major ones.

Noise and vibration can affect the reliability, not only of the drive system itself, but also of the other aircraft subsystems. Hence, every effort should be made to make the gearbox as quiet and vibration free as possible. This may be accomplished by using helical or high contact ratio spur gears, instead of conventional spur gears, by making certain that critical speeds are avoided and by avoiding gear clash frequencies which may reinforce each other.

Another important consideration in the design of any drive system is the efficient use of weight. In general, overdesign means higher reliability, but in aircraft the importance of weight is such that overdesign of some components will invariably lead to underdesign elsewhere. Given the experience with past drive systems, bearing life should <u>never</u> be sacrificed in a weight reduction effort, since bearings are likely to be the main drivers in the establishment of the MTBR. In the basic design of the drive system, it should be realized that, in general, the greater the ratio in the final reduction stage, the lighter the transmission. The lightest freewheel unit design will be possible when the freewheel unit is located on the first reduction stage input where the speed is highest and torque is lowest. By taking advantage of such concepts, the weight allowance of the transmission can be efficiently utilized in improving its reliability.

FUNCTIONAL GROUP ACTIVITY

The functional group charged with drive system design can contribute greatly to the reliability and maintainability of future designs by setting up and continuously updating a problem file. This file, which could be arranged by the various generic components, would essentially be a history of all of the developmental and field problems associated with past and present drive systems. The information contained in this file would consist of descriptions, causes, and solutions of the various problems. The primary purpose of this file would be to prevent past mistakes from being repeated. It would also serve to centralize that information, which would otherwise be

scattered in various memos or in the files of the engineers who worked on the past problems. This procedure also documents "experience" and keeps within the company information that othewise might be lost when experienced engineers leave.

In conjunction with the "problem" file, it would be wise for the functional section to maintain a design manual that covers all aspects of drive system design. This manual should be kept updated with the latest analytical techniques and should include, when appropriate, information that can be gleaned from the problem file. The maintenance of such documents as those suggested here will ensure a uniform and orderly approach to drive system design and will certainly lead to the development of drive systems with improved reliability.

ACCELERATED TESTING

Accelerated testing, i.e., testing at loads greater than those experienced in normal operation, has long been used in the aerospace industry during aircraft development programs. The purpose of such testing is twofold. First, the structural adequacy of the component or system can be substantiated from the standpoint of flight safety. Second, accelerated testing can greatly reduce the test time and therefore the cost of a test program, especially where failure modes of a fatigue nature are anticipated. Accelerated testing has long been effectively utilized in this manner in the structural testing of such helicopter components as rotor blades and main rotor shafts.

Recently, a trend has developed where accelerated testing is being more widely applied to entire gearboxes. The acceleration factor of some of these tests is quite high, with the prorated test power often double the mission prorate or higher. This can lead to failures that may be uncharacteristic of a gearbox operating at normal power levels. Since many of these tests can be classified as "must-pass", it is strongly advised that during design the probability of passing such a test be calculated. It may be found that the design criteria may have to be more severe than the mission in order to have an acceptably high probability of passing the test.

There are a number of components whose performance can be drastically affected in an overstress environment. The most obvious is the drastic reduction in bearing life. Since bearing life varies inversely with the 3.33 power of the load, doubling of the prorated load will cut the bearing life by a factor of 10. Although the chances of spalling a single individual bearing remain small, the probability of spalling a bearing when considering all the bearings in the gearbox can be quite large. A relatively recent example showed that in a 200-hour overstress test, the probability of spalling a bearing was over 25 percent. If the same 200 hours were run at the mission spectrum the probability of spalling a bearing would be less than 2 percent. Considering the possible repercussions of a spalled bearing in a must-pass test, the 25 percent chance of failure is very high.

There are other affects of overstress testing, which, while not as crucial

as bearing spalling, nevertheless can adversely effect the condition of gearbox components. Fretting is more likely, for example, on bolted gear flanges. Bearings are far more likely to creep or spin on shafts. Special care should be given to proper development of bevel gear patterns, since high overloads can drastically shift the patterns, making tooth breakage a more likely occurrence. With spur gears the possibility of hard lines or scuffing increases with the increased load. The tip relief may be insufficient to account for the increased deflection under load. Hence, the teeth go into mesh early and out of mesh late. Under certain circumstances this could lead to contact below the TIF diameter of the gear, i.e., non-involute contact.

As can be seen from the above discussion, there are a number of factors to be considered when a greatly accelerated test is included in a development program. It would be wise to assess the risk of failing such a test as early as possible in the design so that a decision could be made on whether the risk is acceptable or design changes should be made. It is recommended that overstress qualification tests be limited to a prorate of 110 percent of maximum power with only brief durations of running in excess of 120 percent. It is also strongly recommended that those discrepancies which result in disqualification be clearly defined before the test.

BEARINGS

From the standpoint of reliability, bearings are by far the most important gearbox components, since they are among the few components that are designed for a finite life. Bearing life is usually calculated using the Lundberg-Palmgren method. This method is a statistical technique based on the subsurface initiation of fatigue cracks in a through-hardened air-melt bearing material. The life generally used in the B_{10} life, which is the number of hours at a given load that 90 percent of a set of apparently identical bearings will complete or exceed. Over the years, bearing specialists have devised a number of factors that can be applied to B_{10} life so that it more accurately correlates with the observed life. These include material, processing, lubrication film thickness (EHD), speed, and misalignment factors.

REQUIRED BEARING LIFE

The required bearing life depends on the desired reliability of the gearbox (usually stated in terms of MTBF) and the number of bearings in the gearbox. This discussion relates to the MTBF or system life of the bearings alone. The effect of other components on gearbox reliability is presented in the Hazard Function Analysis section. The following expression gives the approximate life for which each bearing should be designed to achieve a given gearbox B_{10} life or MTBF for a system containing N bearings.

$$L_{10} = \frac{N^{.9}\ \text{MTBF}}{5.45}$$

or

$$L_{10} = N^{.9}\ L_{sys}$$

Where L_{10} = design B_{10} life for each individual bearing

N = number of bearings in system

L_{sys} = design B_{10} life of the system

MTBF = design Mean Time Between Failure for bearings alone

The above expression assumes a Weibull failure distribution with a shape parameter equal to 10/9, the value generally used for rolling element bearings. The life (L_{10}) given in the above expression includes lubrication, misalignment, and material factors. Since the misalignment and lubrication factors are usually approximately equal to 1.0, the desired unmodified calculated life, which is generally used in the early stages of a design, can be found by simply dividing the life found in the above expression by the material factor. Typical material factors used for vacuum-melt aircraft quality bearings are 6.0 for ball bearings and 4.0 for roller and tapered roller bearings.

It would be advisable in initial bearing life calculations to use double the required unmodified calculated life for low-speed bearings under 1000 rpm, since the lubrication factor for these bearings is likely to be quite

low.

BALL BEARINGS

Ball bearings are generally used where there is likely to be excessive misalignment or shaft deflection. They are also used, especially in duplex arrangements, where accurate axial positioning is required in the presence of thrust load, such as with bevel gear shafts. Ball bearings are not as common in the main drive train of more recent designs because of advancements made with tapered roller bearings. Ball bearings are, however, often used on lightly loaded accessory shafts. Higher bearing life is easily achieved in these applications due to the very small loads and installation is simplified, since a ball bearing is nonseparable and requires no special setup procedures.

CYLINDRICAL ROLLER BEARINGS

Cylindrical roller bearings are used to support pure radial loads. They are often used at one end of highly loaded gear shafts with either tapered roller bearings or multiple-row matched ball bearings at the other end. Roller bearing life is drastically reduced by excessive misalignment or deflection; hence, when using roller bearings, the stackup of tolerances contributing to misalignment and the shaft or housing deflections should be carefully considered. This is especially true with main rotor drive shafts where high shaft deflection is caused by the rotor head moment. To compensate for some degree of misalignment or deflection and to carry heavy radial loads, roller bearings are crowned to prevent the phenomenon known as end loading (see Figure 2).

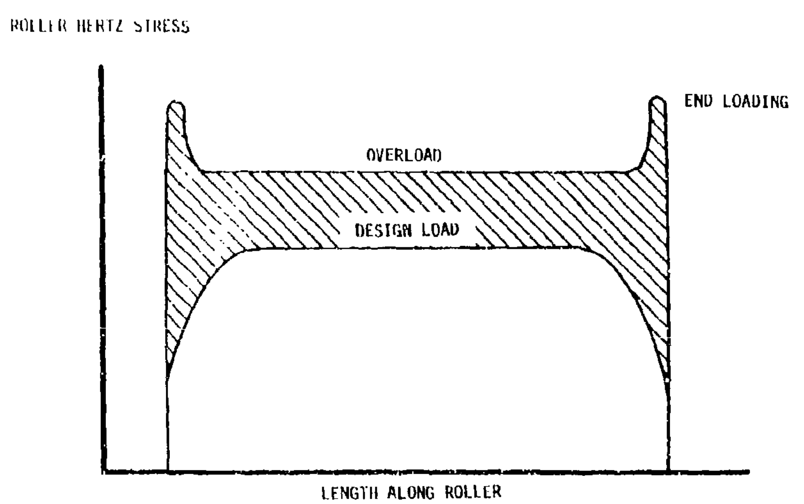


Figure 2. END LOADING OF CYLINDRICAL ROLLER BEARING

End loading invariably leads to a drastic reduction in bearing life. The amount of crown to be used should be based on maximum continuous power. At lower powers, the crown will not significantly change bearing life, while higher powers are usually transients that load the bearing for short durations only.

INTERNAL CLEARANCE

Internal clearance is an important consideration in the design of ball and roller bearings, since improperly mounted internal clearance can drastically shorten the life of a bearing. Too little internal clearance limits the amount of misalignment that can be tolerated and can lead to heavily preloaded bearings, particularly at low temperatures. Excessive internal clearance will cause the load to be carried by too few rolling elements. The best practice is to ensure that under all conditions there will be a small positive internal clearance. Usually, the most significant factors to consider when determining mounted internal clearance of the bearing are the reduction of internal clearance due to shaft or housing fits and the effect of temperautre on the magnesium housing/outer race interface diameters. Given the high press fits generally used with helicopter transmission bearings and the low temperature requirements, it will be found that standard catalog internal clearances are usually not correct.

TAPERED ROLLER BEARINGS

Tapered roller bearings are being used increasingly in helicopter drive systems, since they can react both thrust and radial loads and can offer the greatest load carrying capacity in the smallest possible envelope. Although early tapered roller bearings were speed limited, these restrictions have been removed by utilizing bearings with special lubrication features. However, on very high-speed shafts, the use of tapered roller bearings may be precluded due to their inability to operate for required time intervals under survivability (oil-off) conditions. Tapered roller bearings, unlike single-row ball and cylindrical roller bearings, require spacers or shims to give these bearings the proper amount of preload or end play for proper operation. Usually it is desirable to have a light preload although a small amount of end play is often acceptable. As with internal clearance, extremes in end play or preload should be scrupulously avoided.

PREVENTING BEARING RACE CREEP

The creeping or spinning of bearing inner races on gearshafts is a fairly common, although not usually serious, problem in helicopter drive systems.

Lundberg and Palmgren developed fairly simple parametric calculations for the minimum fit to prevent creep with solid shafts, but there has been little if anything published on minimum press fits for hollow shafts, the type exclusively used in helicopter drive systems. Since an accurate mathematical solution to such a problem would be extremely difficult, the best approach at this time seems to be a reliance on past experience.

Sometimes it may not be possible to achieve the necessary press fit to prevent creep without introducing excessively high hoop stress in the bearing race. A common practice in this case is to use separate antirotation devices with a slotted bearing race. Although this practice is fairly effective with stationary races, it is seldom effective with rotating races. It is better in cases like this to use special bearings with thicker or wider inner races, as shown in Figure 3, to get the desired antirotational force.

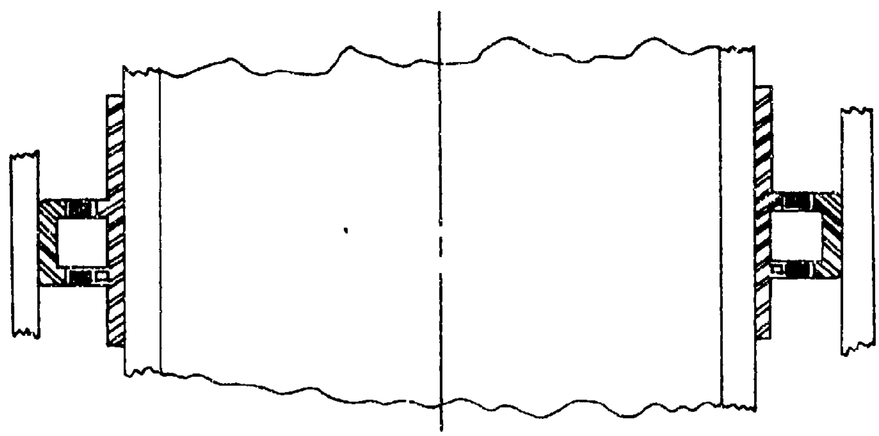


Figure 3. ROLLER BEARING WITH EXTENDED INNER RACE

BEARING MATERIAL

By far the greatest advance in bearing technology has been the development of extremely clean bearing steels resulting from vacuum-melt processing. Vacuum-melt 52100 bearings, for example, offer one and one-half to two times the life of vacuum-degassed 52100 bearings. Bearings of such advanced materials as M-50 steel can offer even further improvement. Just because these materials are available, however, does not mean that all bearings have to be made of vacuum-melt material. Depending on the bearing, vacuum-melt bearings can be from 2 to 5 times more costly than vacuum-degassed bearings, as well as require longer lead times to procure. Hence, the following policy should be used for bearing material selection. All high-speed primary power train bearings should be fabricated of M-50 (or in the case of tapered roller bearings, carburized CVM steel) for the sake of improved survivability following loss of lubrication. All other bearings should be fabricated of vacuum-degassed material unless there is a problem with low life, in which case the more expensive vacuum-melt materials should be used.

INSPECTION REQUIREMENTS

Proper inspection of bearings can significantly reduce their mortality rate. Besides the obvious dimensional inspection requirements, two additional inspections should be specified for all helicopter drive system bearings; magnetic particle and nital etch. Magnetic particle inspection can detect the presence of relatively large surface or near-surface anomalies, such as inclusions, which are often the cause of bearing spalls. Nital etch inspection can detect the presence of grinding burns, which locally change the hardness of the material and also result in premature bearing failure. Recently research has been done on such techniques as magnetic perturbation, laser-scattered light, and Barkhausen noise to accurately detect surface and near-surface anomalies in bearings. These methods are presently used on an experimental basis due to extremely high costs.

INSTALLATION/REMOVAL

The installation of bearings should be carefully considered during design not only to prevent assembly errors, but also to permit easy removal of the bearing without damaging it. Lead chamfers should be provided at all bearing journals to facilitate installation. When specifying the breakout on the bearing corners, the shaft drawing should be checked to ensure that the maximum radius at the shaft shoulder will be cleared by the bearing. The height of the shaft shoulder should, if possible, be consistent with that recommended by bearing manufacturers. Where necessary, flats should be machined on the shaft shoulder so that a bearing puller can remove the bearing by contacting the inner race. Many bearings have been damaged in the past where the bearing puller could grab only the cage or rollers of the bearing. Where duplex bearings are used, the bearings should be marked so that the installer can readily determine the proper way for the bearings to be installed. Incorrectly installed duplex bearings will not properly react the design loads. All bearings that can be separated should have the serial number clearly shown on all of the separable components. This will prevent the inadvertent mixing of components. Every assembly drawing that contains bearings should clearly explain in the drawing notes how the bearing should be installed. It is imperative that the mechanic building up an assembly have this information readily available.

TAIL ROTOR DRIVESHAFT BEARINGS

Tail rotor driveshaft bearings present a slightly different problem to the drive system designer than bearings within the gearboxes. Load capacity is rarely, if ever, a problem since the only loads on the bearing are those due to a shaft weight, imbalance, and misalignment. Misalignment is usually not a problem with bearings mounted within viscous dampers, which in present-day drive systems is usually the case, If, however, a hard-mounted bearing is used, provision for the misalignment must be made either by using high internal clearance or by some sort of spherical outside diameter that will allow the bearing to move as a complete unit to compensate for the misalignment.

Contamination of the lubricant either by water or by gritty particles is the most common reason for failure of these bearings. To resist corrosion vacuum-melt stainless steel should be used as the bearing material, since the dynamic capacity of the bearing is not crucial. Another technique that has been developed recently to prevent corrosion is the application of very thin chrome plating to the bearing elements. It has been reported that these coatings have performed very well in test.

There has been some debate as to whether it is desirable to have tail rotor drive shaft bearings with removable seals. The advantage in having removable seals is that the bearings may be inspected at overhaul, and if found in satisfactory condition may be repacked and reused. Unfortunately in the past when removable seals have been used, organizational maintenance personnel have removed the seals and have attempted to repack the bearings themselves. More often than not, they would overpack the bearings with grease, which led to overheating and eventual failure. Generally these bearings should be packed only about one-third full with grease. Hence, if removable seals are to be used, specific instructions should be provided in the technical manuals on the correct way to pack these bearings.

GEARS

Drive system design usually begins with the selection of the gearing that will be used to transmit the power from the engines to the main and tail rotors. The process begins with a layout showing location of engine inputs, main rotor shaft centerline, and tail drive shaft centerline. This geometry, along with the required reduction ratio, dictates to a large extent what combination of spur gears, helical gears, spiral bevel gears, and planetary reduction units will be used to deliver the power to the rotors. It is generally desirable from a weight and size standpoint to have as large a ratio as possible in the final reduction stage.

The balance of this chapter will address itself to the detail design of spur, helical, and spiral bevel gears, with a discussion of gear web design included. A separate section dealing exclusively with the design of planetary units is also provided.

SPUR GEARS

Spur gears are commonly used in helicopter drive systems both for parallel-axis speed reduction and in coaxial planetary units. In general the reliability level of helicopter drivetrain spur gears is extremely high using present design standards, and maintainability is not an important consideration in the design of spur gears. There are, however, some design considerations that will be discussed here because they are sometimes overlooked.

Balancing the Design

Generally the initial design of a spur gear mesh is one of standard proportions and equal tooth thickness for both pinion and gear. This is rarely, however, the optimum configuration for a spur gear mesh because this type of design does not have two very desirable characteristics: recess action, and balanced bending stresses in pinion and gear. A recess-action gear mesh has a long addendum pinion and short addendum gear. In this type of design all or most of the sliding takes place as the gear teeth are moving away from each other. A recess-action mesh is quieter and smoother running than a standard mesh and has a much lower tendency to score due to better lubrication characteristics within the mesh.

Although the advantage of having balanced bending stresses on pinion and gear is primarily lower weight, it does have an indirect bearing on reliability. As was stated earlier, whenever there is an inefficient use of weight, reliability is compromised somewhat. For example, even a fraction of a pound wasted through not optimizing a spur-gear mesh could instead be applied to a bearing where the life could perhaps be doubled. Hence, while the emphasis on light weight can be detrimental to reliability, the carrying of excess weight can also have an adverse effect. High reliability in a helicopter drive system depends largely on the efficient use of weight.

Fortunately it is usually a fairly simple task to achieve recess action and balanced bending stresses with most spur gear meshes. This is accomplished by shifting the length of contact up the line of action towards the driven gear, as shown in Figure 4, while increasing the circular tooth thickness of the pinion and decreasing that of the gear. It is essentially a trial and error procedure but is usually easily done.

Design Criteria

There are four design criteria that are used to evaluate the adequacy of spur or helical gear design: bending stress, hertz stress, flash temperature index, and/or lubrication film thickness. The first three have long been used in gear design and the methods of calculation are well documented in many publications. EHD film thickness is a more recently developed criterion that some gear specialists have advanced as a check for scoring probability. It can be appreciated that if an oil film of a thickness greater than the contacting surface asperities can be maintained, scoring will not occur. Although the exact method for calculating EHD film thickness is not yet standardized, it is highly recommended that it be used as an added check on spur gears, especially those with low pitch-line velocities. There has been at least one instance of chronic pitting, which an EHD film thickness calculation may have prevented. It is interesting to note that the compressive stress in the particular mesh was well within the generally recognized allowable range recommended by American Gear Manufacturers Association.

The subject of allowable tooth stresses is a controversial one and no attempt will be made here to outline specific values. Each company designing gears seems to have its own set of criteria and limits that have evolved over the years, and this kind of experience is difficult to refute.

The same can be said of S-N curve shapes. Experience has shown that the use of allowable stresses published by AGMA will result in satisfactory design and performance.

Profile Modification

In order to insure smooth conjugate action under load, it is generally the practice to modify the involute profile, usually with tip relief, to correct for the deflection of the gear teeth under load. The deflection of the gear teeth under load should be calculated as accurately as possible, since too little or too much relief can both be detrimental to the performance of spur gears. Too little relief causes the gear teeth to go into mesh early and to go out of mesh late. This results in higher dynamic loads with attendant increases in stress, vibration and noise, and possible non-involute contact that can lead to hard lines, scuffing, or scoring of the gear teeth. Too much tip relief lowers the contact ratio of the gear set and again can result in less than optimum performance with respect to stress, vibration, and noise.

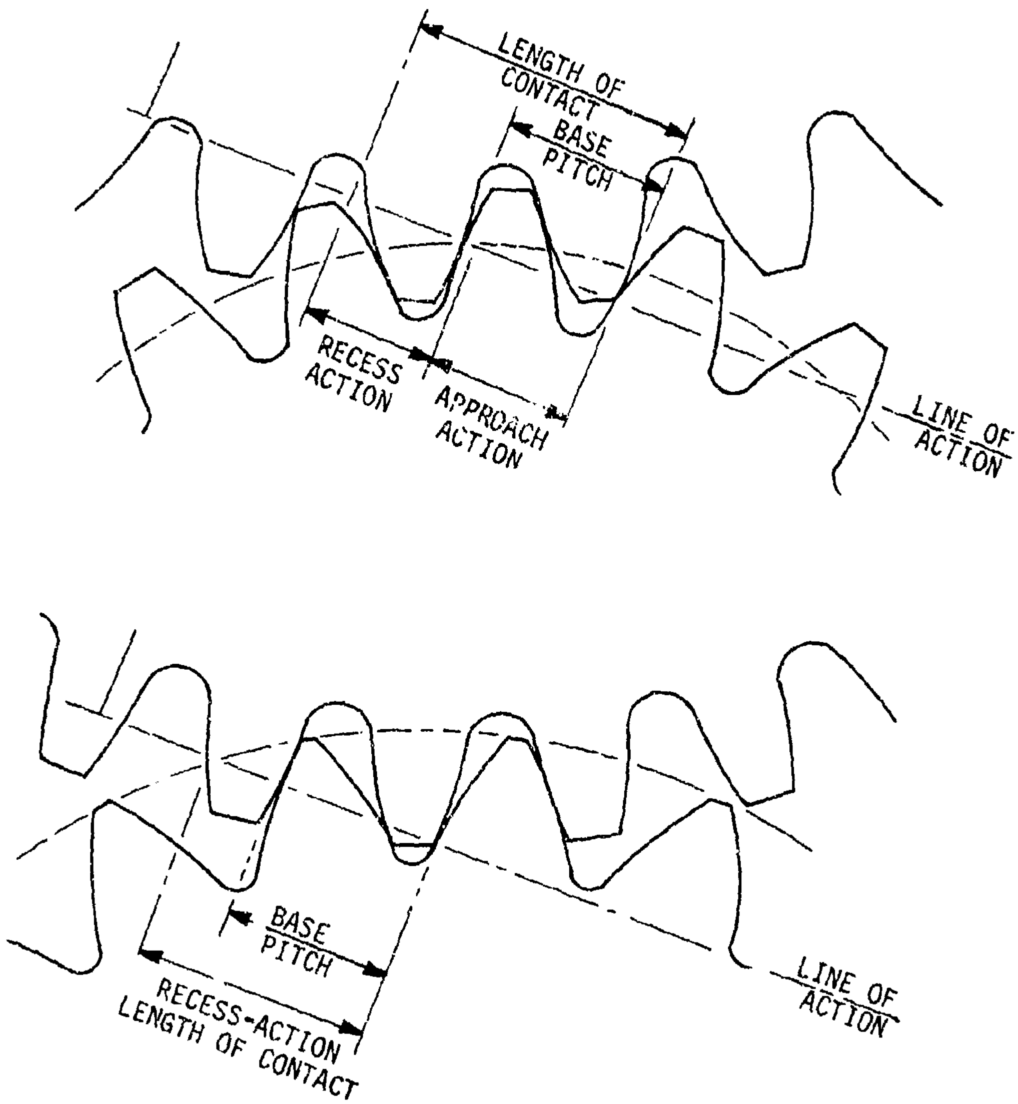


Figure 4. CONVENTIONAL AND RECESS-ACTION SPUR GEAR MESHES

Crowning

Crowning is generally applied to spur gears to insure full contact across the face of the gear without end loading. Again both excessive and insufficient crown can be detrimental to spur gear performance. With insufficient crown, end loading will occur that will result in higher than predicted bending and hertz stresses. With excessive crown the full face width will not be used and again, higher than predicted hertz stress will occur in the center of the tooth. The method used to calculate the amount of crown for a spur gear tooth is very similar to that used to calculate the crown on bearing rollers

Helix Correction

If the bearing mountings at either end of a spur gear are not of equal stiffness the spur gear will tend to cock under load. This will cause the load to be concentrated at one end of the tooth, resulting in increased stresses. This condition is more pronounced with gears with high L/D ratios. By grinding a helix across the face of the gear, this condition can be corrected. Unfortunately analytical prediction of the required helix correction, if any, is not very practical and visual pattern inspection can be misleading. Strain gaging the gear rim is perhaps a better method of determining correct helix corrections. This can be done by iteratively grinding a helix across the face of the gear until the strain gages on either side of the face give equal readings.

When grinding for helix corrections, however, it must be certain that the location of the bearing bores of the housings used during the test are to blueprint tolerances. If the housing is bad, grinding helix correction into gears which are used in good housings could cause the very problem that helix corrections are designed to prevent.

Edge Break

It is advisable that all gear teeth, including spiral bevel and helical gear teeth, have the edges broken. This will prevent chipping of the gear teeth at the corners, which carburization has made very brittle.

HELICAL GEARS

Helical gears are not nearly as common in helicopter drive trains as spur gears, although they are quieter and have greater load-carrying capacity per inch of face width than spur gears. The one disadvantage of helical gears is that a thrust load is introduced along the gearshaft, thereby necessitating somewhat larger bearings. This problem can sometimes be overcome if a spiral bevel gear is attached to the same shaft. In this case, the bevel and helical gears can be designed so that their thrust loads oppose each other. Another way to avoid the thrust problem is to use double helical gears (sometimes called herringbone). Here the thrust loads developed by the opposing helical gear are canceled out. The problem with this type of design is the difficulty of timing between the facing gears.

Analysis of helical gears is very similar to that used for spur gears. The stress analysis is performed using an equivalent spur gear tooth. The strength analysis of helical gears, like that of spurs, is outlined in AGMA standards.

SPIRAL BEVEL GEARS

Spiral bevel gears are found, almost without exception in every present-day helicopter drive system. They are probably the most difficult gears to design, since the geometry of spiral bevel gears is considerably more complicated that that of spur or helical gears. The Gleason Works of Rochester, New York has developed much of the current spiral bevel gear technology, including the analytical techniques used throughout most of the helicopter industry. Although the literature published by Gleason contains much of the information needed to design reliable spiral bevel gears, some points will be covered here.

Basic Design

The hand of spiral of the gears should be chosen if possible, so that the axial force tends to push both pinion and gear out of mesh. If that is not possible, then the hand of spiral should be chosen so that the pinion is forced out of mesh. The face contact ratio of the mesh should be as high as possible to ensure smooth quiet running. While a face contact of 2.0 or greater is preferred, a face contact ratio as low as 1.25 is acceptable. The face width of a spiral bevel gear should not in any case exceed one-third of the outer cone distance or the gear becomes impractical to manufacture because of the very small tooth depth at the toe of the gear and because load concentrations at the toe end could lead to tooth breakage.

Gear Pattern Development

The development of correct tooth contact patterns is extremely important with spiral bevel gears, since incorrect patterns can lead to high bending stresses and eventual tooth fracture. Although gear pattern development is a rather complex subject and largely beyond the scope of this guide, some general points will be brought out. First, in general, a central toe bearing equal to approximately one-half the tooth length, as shown in Figure 5, is specified under light load conditions. In practice the tooth bearing pattern nearly always spreads out and shifts toward the heel under full load, so that the operating pattern will appear as shown in Figure 6. It is recommended that a V and H check be performed on all spiral bevel gears, since that is a more accurate method of determining the required length of bearing than merely visually inspecting the patterns. The V and H check is a method for measuring the amount and direction of the vertical and axial displacements of the pinion from its standard position to obtain tooth bearing patterns at the extreme toe and the extreme heel of the tooth. The V and H readings are then compared to the readings of a master gear set. When the total vertical movement of the V and H check is too large, it indicates that the tooth bearing is too short, and therefore the load will be concentrated on too small an area of the tooth surface, which could lead to

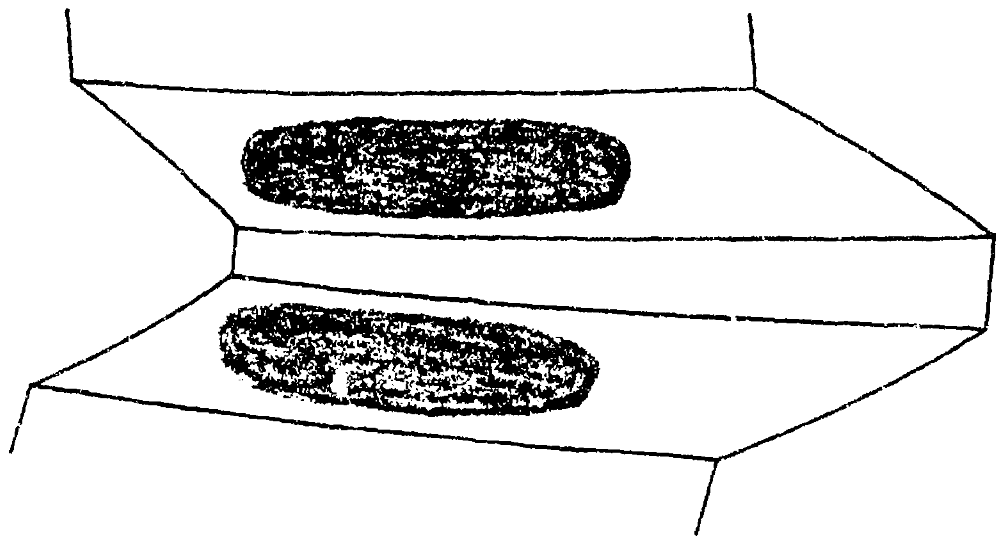


Figure 5. SPIRAL BEVEL GEAR CENTRAL TOE BEARING PATTERN

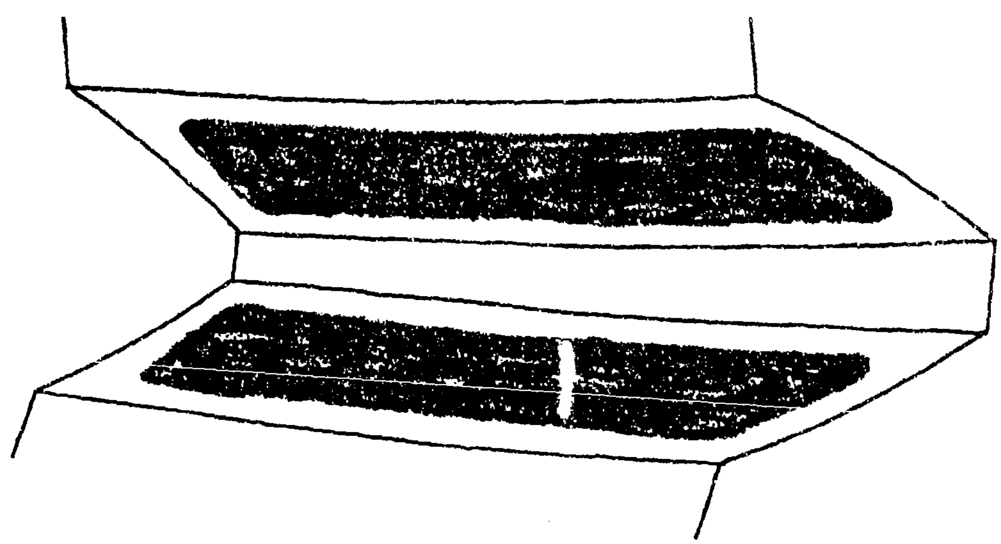


Figure 6. PREFERRED SPIRAL BEVEL GEAR BEARING PATTERN UNDER FULL LOAD

excessive wear. When the total vertical movement is too small, it indicates that the tooth bearing is too long, and hence, the gears will lack sufficient adjustability to compensate for mounting deflections, which may lead to load concentration on the ends of the teeth. Acceptable tolerances of the desired V and H check readings should be provided to quality control by the gear design engineer.

Fillet Radius

Unlike with spur gears or helical gears where the root fillet radius can easily be checked with conventional radius gages, the checking of the fillet radius of spiral bevel gears is extremely difficult with conventional methods. It is recommended that epoxy casts of the tooth spaces of spiral bevel gears be taken to facilitate the inspection of this extremely important feature. Figure 7 shows some of these casts.

Figure 7. SPIRAL BEVEL GEAR TOOTH INSPECTION CASTS

GEAR WEB AND RIM DESIGN

Although in the past web fracture has not been a serious problem with drive system gears, the design of the web should be given careful attention, since failures of gear webs are extremely serious. When designing gear webs, it is desirable that the web be centrally located with respect to the face and lined up with the direction of load on the gear. Webs located off center with respect to the face are to be avoided if at all possible.

Care should be taken with spur gear webs so that they are not designed too thin. When analyzing such webs, assume a trapezoidal or triangular load distribution across the face of the gear, rather than a uniform load distribution, as shown in Figure 8. This is especially true if the gear web is attached to the shaft flange through a bolt circle.

Care must also be taken that the gear rim is not undersized. Since accurate analysis of gear rims by other than finite-element programs is difficult, it is best to use a conservative approach in selecting gear rim thickness. Experience has shown that a gear rim equal to 1.15 to 1.20 times the whole depth of the tooth is a good rule of thumb.

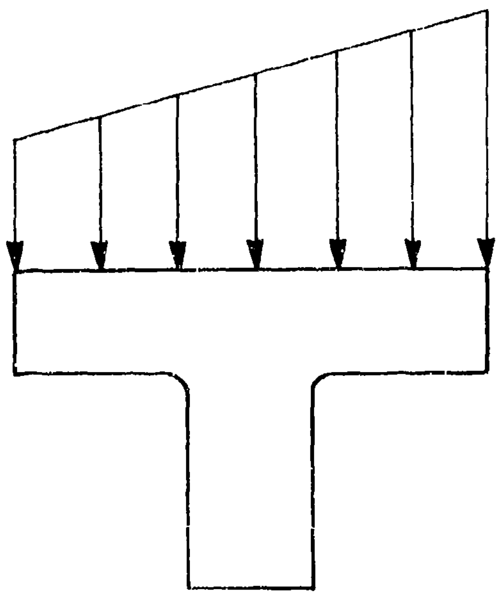


Figure 8. ASSUMED LOAD DISTRIBUTION FOR ANALYSIS OF THIN SPUR GEAR WEB

PLANETARY UNITS

Planetary units are used in many helicopter drive systems because they offer relatively large speed reductions in a compact package. Because the load is shared among the pinions, the face width of the planetary gears is much less than that which would be needed with a single mesh reduction.

There are several points to remember in the basic design of a planetary Unit. First, it is desirable from a weight standpoint to use as many pinions as possible in the unit. Sometimes it is possible to add another pinion through the use of stub teeth on the pinions. When choosing a planetary design, it is desirable not to have the equally spaced planets meshing in unison with the sun or ring gear. This may be assured by designing the planetary unit so that the following relationship holds:

$$\frac{Nr}{n} \text{ or } \frac{Ns}{n} = k + \frac{a}{n}$$

Where Nr = number of teeth in ring gear

Ns = number of teeth in sun gear

n = number of planet pinions

k = a whole number

$\frac{a}{n}$ = an irreducible proper fraction

The analysis of planetary gears is exactly the same as for normal gears except that with the reverse bending experienced by the pinion teeth, the allowable bending stress is lower. Often it will be necessary to grind helix corrections across the face of the sun and ring gears, especially if a roller/thrustwasher-type support is used for the pinions. The amount of this correction can be approximated during design by calculating the slope of the planetary plates.

There are two basic types of pinion bearing supports that may be used in planetary units: roller/thrustwasher and spherical bearings. These are shown in Figures 9 and 10, respectively. The roller/thrustwasher design is generally used for high-power planetary units where both an upper and lower plate are needed to support the planet posts. The most common problem with this type of design is thrustwasher wear. The excessive wear generally results from an inadequate supply of lubricant to the thrustwasher area. Hence, special attention should be given to this area with respect to lubrication. The spherical bearing-type support is generally the preferred design from a reliability standpoint, since there are fewer parts and the thrustwasher problem is eliminated. The spherical bearings also allow the pinions to maintain alignment with the sun and ring gears despite deflection of the pinion posts. This obviates the need for any helix correction on the sun and ring gears. Despite the advantages of this design, it may be impossible with very high torque planetary units to provide

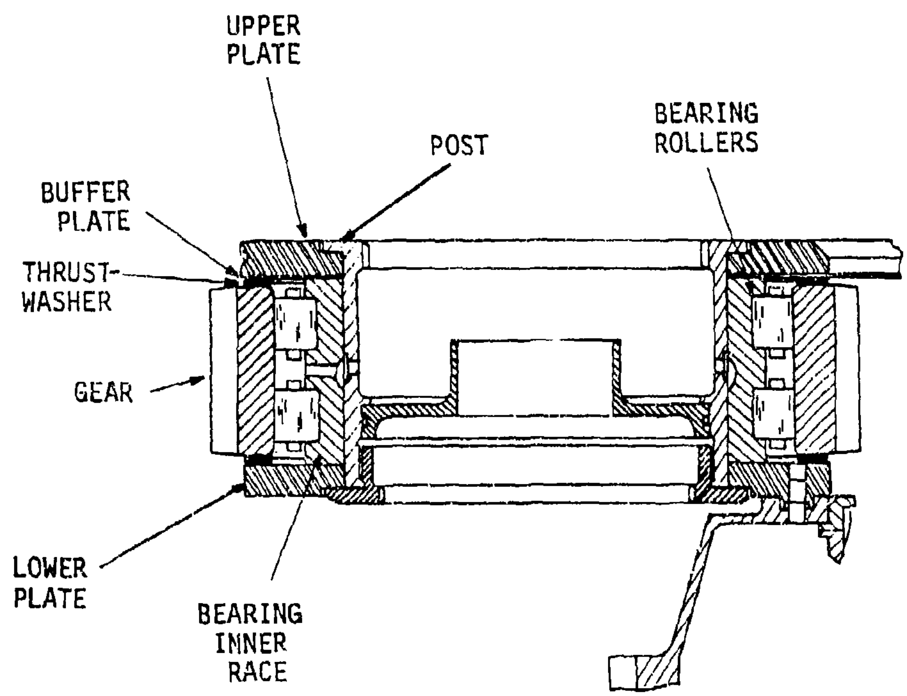


Figure 9. ROLLER/THRUSTWASHER PLANETARY PINION SUPPORT

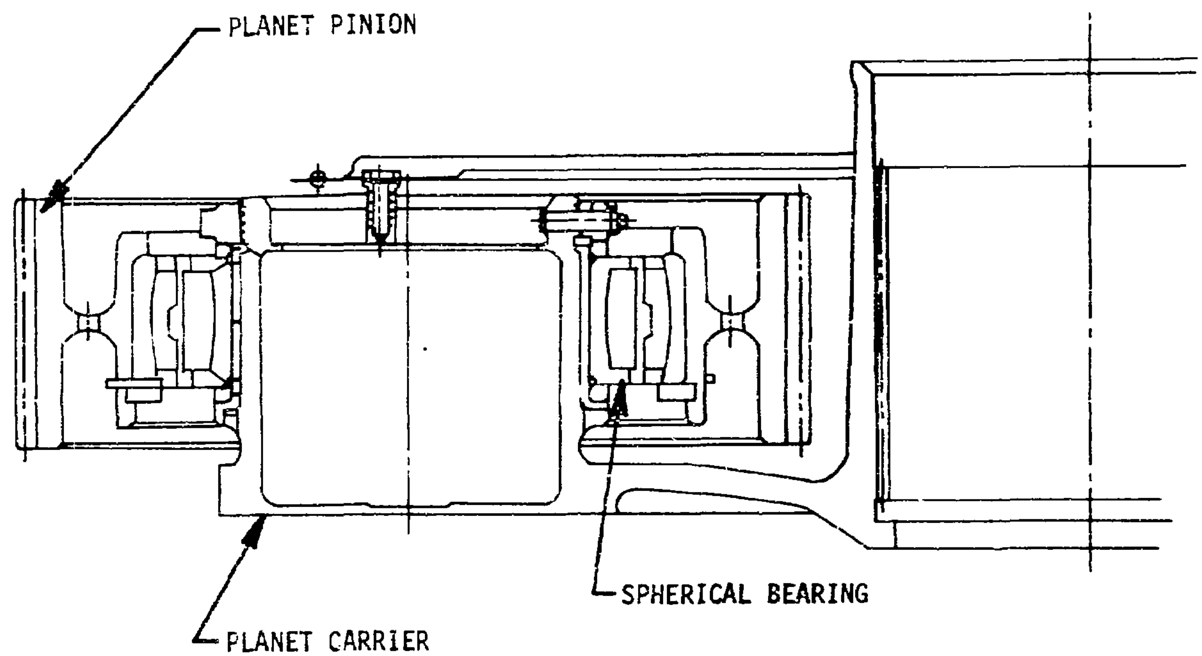


Figure 10. SPHERICAL BEARING PLANETARY PINION SUPPORT

adequate support to the pinions with a cantilever design. In such cases it will be necessary to use the two-plate design.

OVERRUNNING CLUTCHES

There are three basic types of overrunning clutches suitable for application to helicopter drive systems: sprag, spring, and ramp roller. These are shown in Figures 11 through 13.

The sprag type is by far the most common. The ramp roller has been used exclusively by only one airframe manufacturer. The spring clutch, which has shown promise in some research and development programs, has yet to be used on a production aircraft. The detail design of all three types of clutches is the subject of a design guide recently published by the Applied Technology Laboratory, U. S. Army Research and Technology Laboratories (AVRADCOM).

The most common reliability problem with all three types is excessive wear at the overrunning interfaces. This problem has been especially frequent with auxiliary power unit clutches, which spend far more time than primary power train clutches in the overrunning mode. The problem is extremely difficult to deal with during design. Wear rates of the clutches are hard to predict and vary widely depending on the aircraft, operator, and clutch location. Even if wear rates could be predicted, the clutch designer cannot add excess material to account for wear because of the extremely close tolerances required for proper operation of the clutch.

Lubrication of an overrunning clutch is obviously very critical, since it can have a large impact in determining the life of the clutch. It is generally desirable for the sake of lubrication to have the inner member of the clutch as the output member. This simplifies lubrication during overrunning, since the centrifugal force may be used to force the lubricant to the sliding parts of the clutch through radial holes in the inner member. If the inner member is the driving member, the entire cavity of the inner member must be pressurized with lubricant to provide sufficient lubricant during overrunning.

It is best from a size and weight standpoint for overrunning clutches to be located at the earliest possible reduction stage, preferably at the transmission input. While all three types of clutches have operated successfully up to 20,000 rpm in test programs, there are differences in speed capability among the three. The ramp roller clutch, because of its sensitivity to centrifugal effects, should perhaps be limited to 12,000 rpm. The sprag clutch is probably at its limit at 20,000 rpm. However, the spring clutch, because of its simple design, could be applied at speeds greater than 20,000 rpm. All clutches that are to be used at speeds greater than 6,000 rpm should be dynamically balanced.

Special attention should be paid to certain design details of the various clutches to ensure reliable operation. For the spring clutch, concentricity between spring, arbor, input bore, and output bore is critical. A path should be available for the lubricant to flow along the entire length

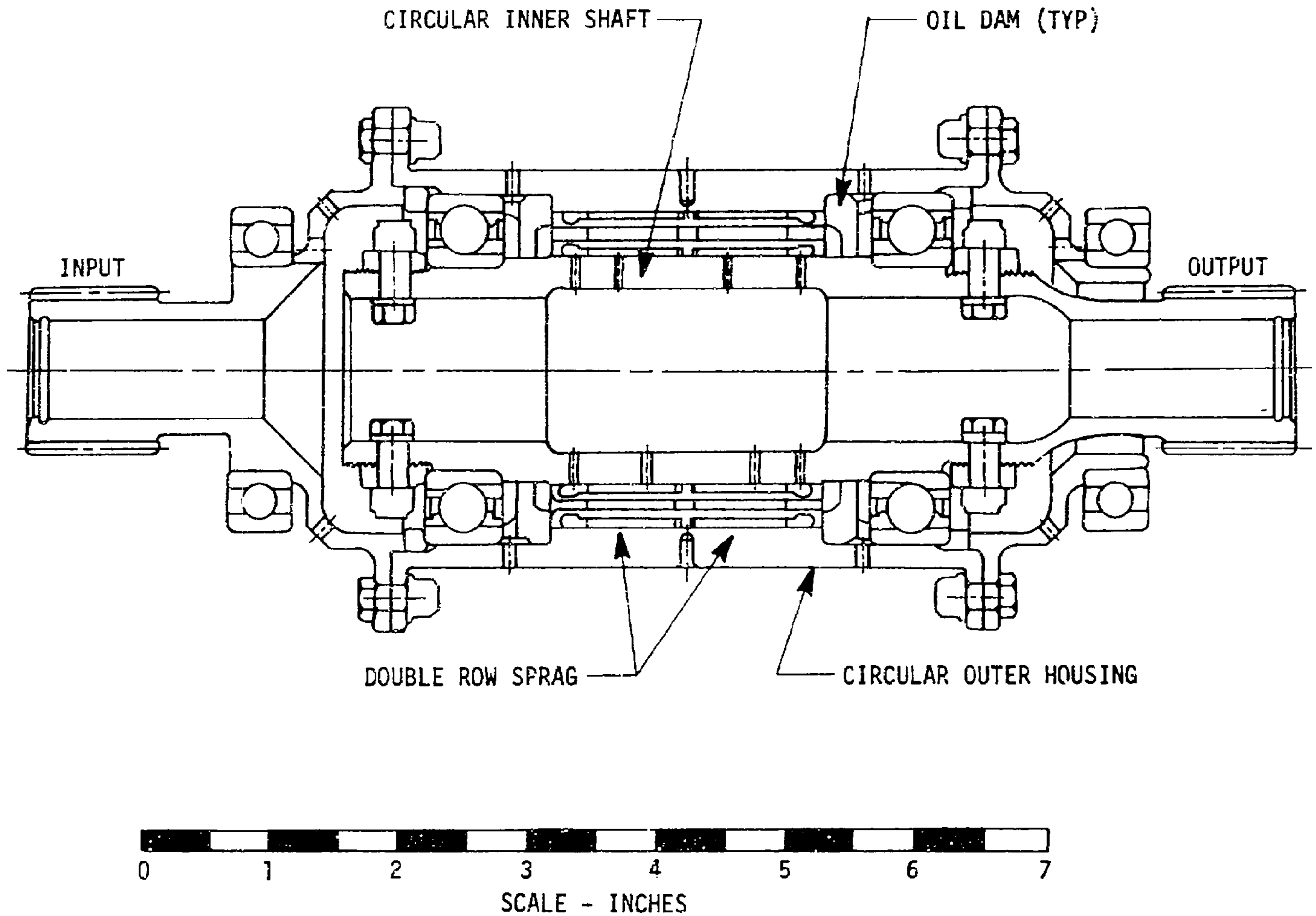


Figure 11. SPRAG-TYPE OVERRUNNING CLUTCH

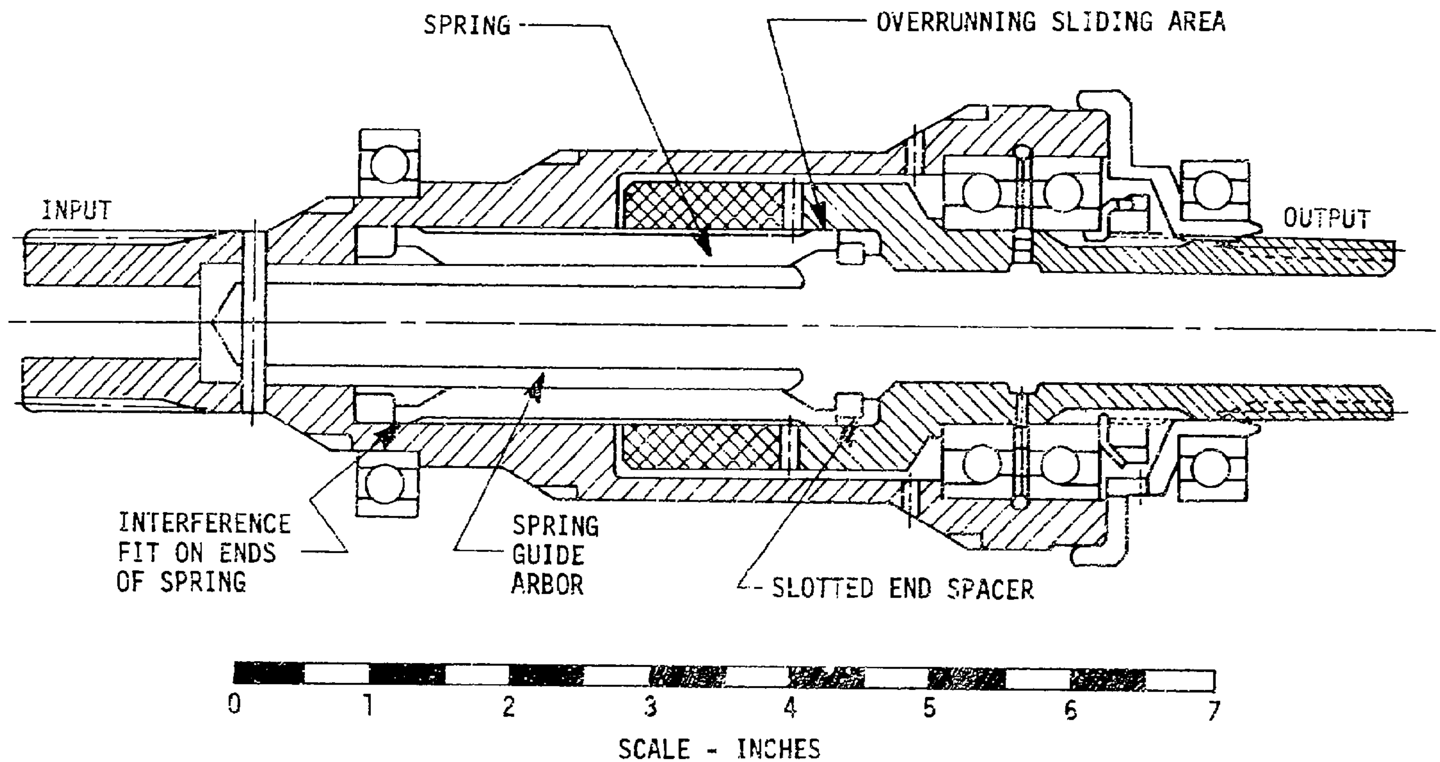


Figure 12. SPRING-TYPE OVERRUNNING CLUTCH

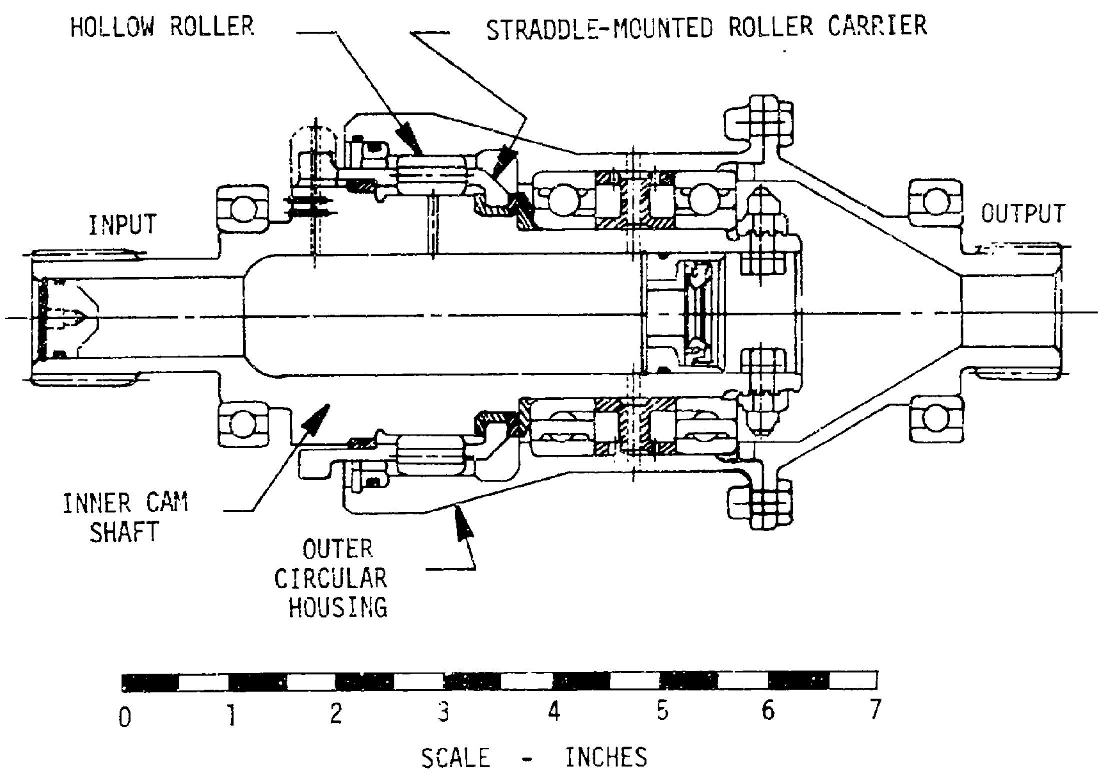


Figure 13. RAMP ROLLER-TYPE OVERRUNNING CLUTCH

of the housing/spring interface. This may be accomplished by slots in the spring or housing. The overrunning coils at the ends of the spring should be free of any type of plating.

For the sprag clutch, oil drainage should be such that when the clutch is at rest all oil is drained from the sprag area. At low temperatures, the thickened lubricant remaining in the sprag cavity could cause the sprags to hang up momentarily and lead to shock loading of the clutch and possible sprag fracture. The clutch diameter should be as small as possible to limit centrifugal effects, and yet be consistent with permissible stress levels. For high torque applications, it may be better to have a small-diameter double-row sprag unit rather than a larger diameter single-row sprag unit. To insure proper loading of the sprag units, the clutch bearings should straddle the sprag unit.

For the ramp roller clutch, full accounting must be made of the effects of centrifugal force on the cage and pin/spring mechanism. Again, clutch diameter should be designed to be as small as possible to limit the centrifugal effects on clutch operation.

It appears from a reliability standpoint that the spring-type overrunning clutch is superior to both the sprag and ramp roller clutches. Its simple design with very few parts makes it an attractive alternative to the more common, older types, and it should be seriously considered for use in future drive systems.

COUPLINGS

Although early couplings, which required periodic lubrication, have been the source of considerable reliability difficulties, these problems have been essentially eliminated with the widespread use of the disc pack or Thomas coupling, shown in Figure 14. This coupling, which is used primarily in applications requiring very little relative axial displacement and low misalignment of about .5 degrees continuous, has an excellent service record. The coupling is simple, lightweight, and maintenance free. It is also essentially immune to handling damage and corrosion. The most common failure mode of Thomas couplings is fretting of the steel laminates around the bolt holes. Since analysis of this is very difficult, the problem can be mitigated by use of a high bolt preload and a generously large bolt-circle diameter.

The KAflex^R coupling, shown in Figure 15, appears to be the preferred choice from a reliability standpoint, when the coupling must withstand continuous misalignments of 3 degrees and transient misalignments up to 6 degrees. The KAflex^R coupling, unlike the Thomas coupling, will also withstand axial deflections. Requirements such as this may become much more common, if transmission isolation is adopted to a greater extent to reduce cabin vibration. Like the Thomas coupling, the KAflex^R coupling is simple and maintenance free. Although not yet in production (some UH-1's and AH-1's will be retrofitted with these couplings in the near future), KAflex^R couplings have seen over 6,000 hours of flight time on UH-1 and AH-1 aircraft with no apparent problems.

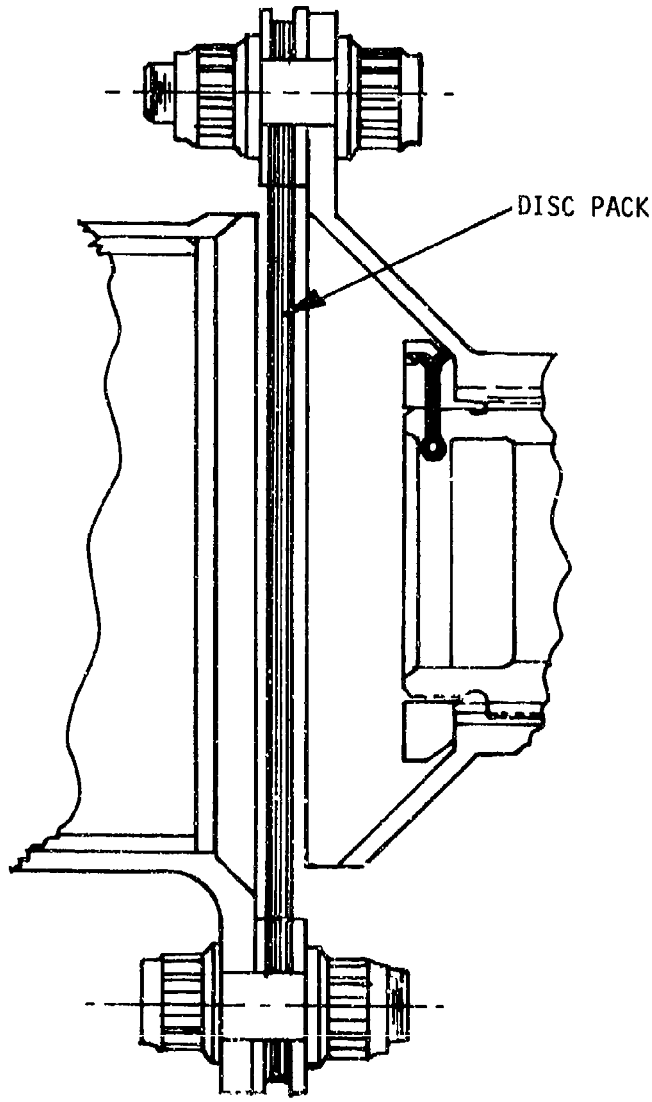


Figure 14. DISC PACK THOMAS COUPLING

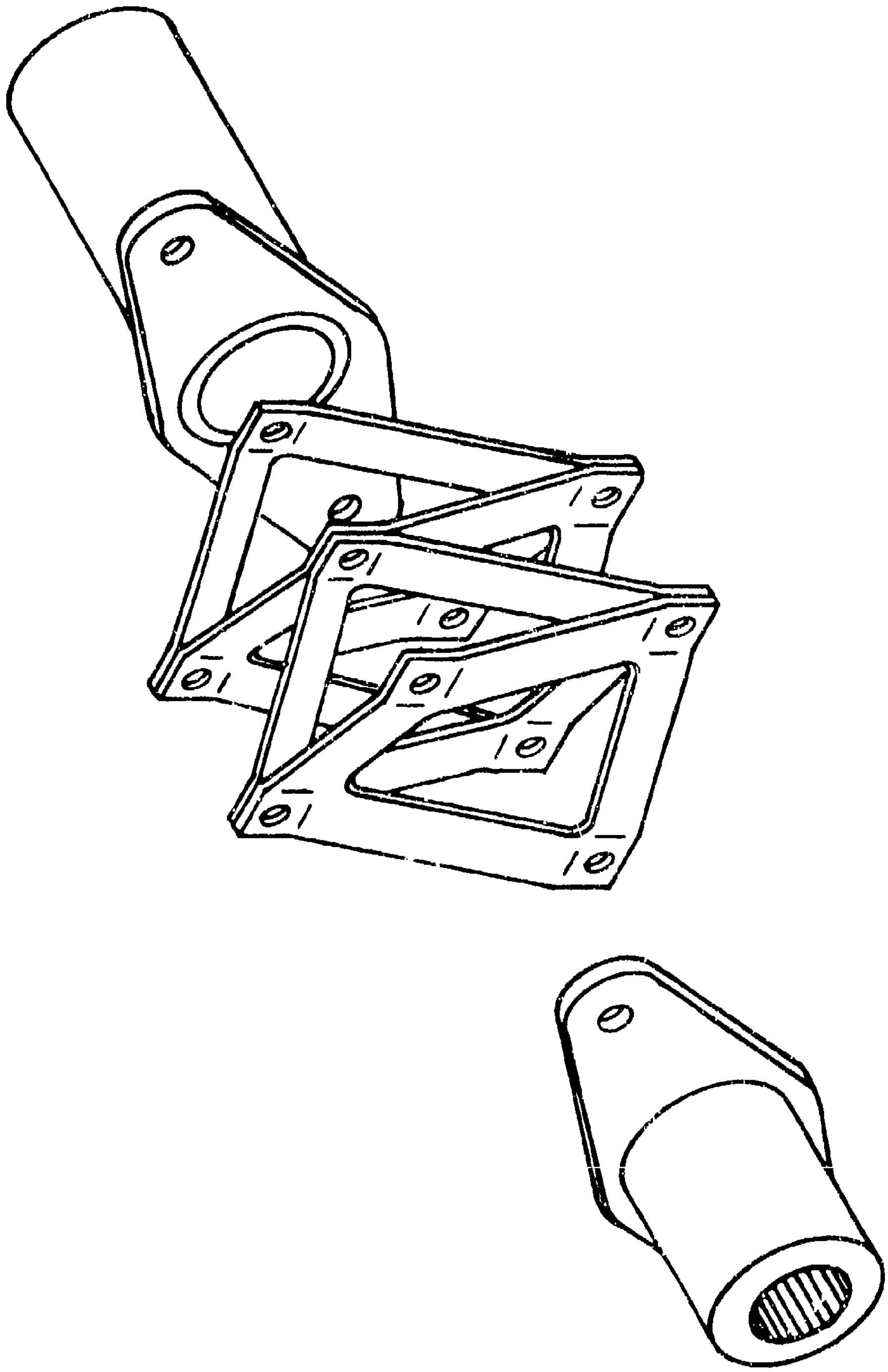


Figure 15. KAFLEX® COUPLING

Gear-type couplings, shown in Figure 16, have been used exclusively when a high misalignment capacity of 3 degrees continuous, 6 degrees transient is required. This type of coupling will also allow substantial axial motion. Unlike the two previous types discussed, the gear-type coupling requires lubrication, and if it is not located within the gearbox housing where it can be lubricated by the gearbox lubrication system, it will require periodic maintenance. Gear couplings are also considerably heavier than the other types discussed.

SHAFTS

Shafting has traditionally not been a large problem in helicopter drive systems. There are relatively few possible failure modes, and these occur extremely rarely. Most discrepant driveshafts can be attributed to maintenance errors, or mission-related damage, such as ballistic strikes. It is advisable to keep shafts covered to prevent damage from flying debris.

Another consideration in the design of shafts is a dynamic balance requirement for all high-speed shafts. This is important for all rotating shafting including gearshafts within the gearbox. A good rule of thumb is to balance all shafts over 6,000 rpm so that the imbalance force is less than 10 pounds.

ROTOR BRAKES

The rotor brake on military and commercial helicopters is used primarily to stop the rotor or rotors from full or partial rpm within a specified time with the engines either at ground idle or shutdown. It is also used to hold the rotor stationary during some ground operations with the engines idling to minimize the risk to ground or deck personnel. Other reasons for a rotor brake include quick dispersal and rapid concealment during shipboard operation, and emergency stopping when fast egress is required. The normal stopping time usually consists of 5- to 10-second delay, during which the rotor rpm decays naturally due to windage and drag, plus the actual braking time. The brake should also be capable of actuation at 100 percent rotor speed without overheating.

On large- and medium-sized helicopters, the rotor brake is a hydraulically actuated disc brake where lining wear is compensated for by a variable-displacement feature that increases the volume of brake fluid or by a mechanical device which maintains a constant clearance between lining and disc as the linings wear. The brake actuation system can either be a full-on/full-off system that eliminates the pilot from control once he activates the brake, or it can be a controlled magnitude system that allows the pilot to control the rate of deceleration by varying the input force or brake lever position.

In addition to developing sufficient friction torque to stop and/or hold the rotor, the brake disc must also act as a heat sink and be capable of absorbing and dissipating the kinetic energy of the rotors and other rotating parts. Enough volume of material must be provided in the disc to

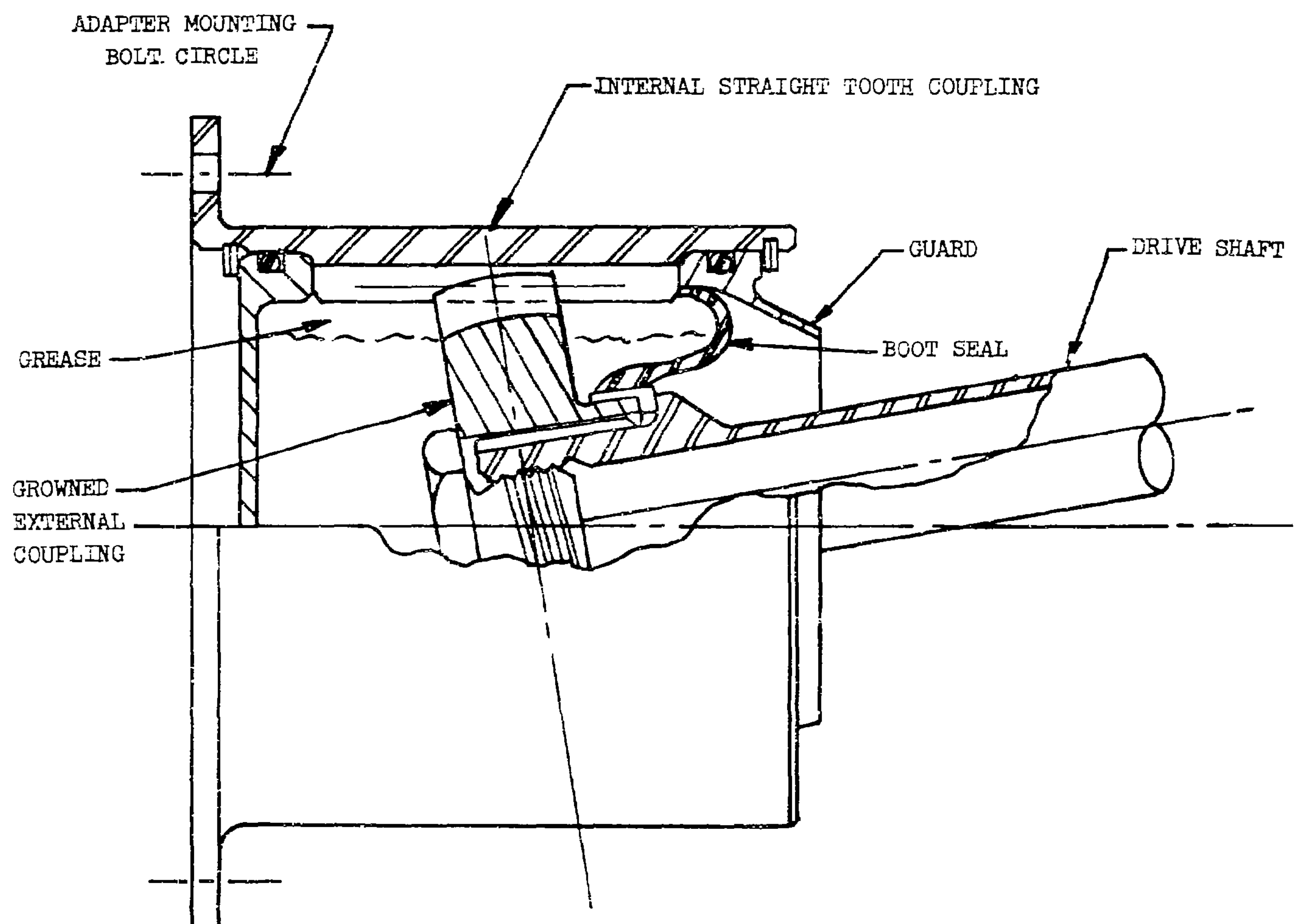


Figure 16. GEAR COUPLING

keep the temperatures within the thermal limits of the material and to prevent disc warpage.

Certain safety features are required to prevent inadvertent brake actuation in flight. These may take the form of mechanical or electrical interlocks that must be deactivated before brake actuation. A relief valve and/or mechanical stop should be included to insure brake operation at a safe load level. The location of the disc in the drive system should be carefully considered not only to provide an adequate torque input to the rotor but also to isolate the brake from flammable fluids or materials. Care should be taken to assure adequate operating clearance between lining and disc to preclude the possibility of drag, thus producing a continuous source of heat in flight that could ultimately result in a fire.

Standard mechanical engineering design principles for brakes and clutches also apply to rotor brakes. Numerous texts describe proper analysis methods. Static and dynamic friction coefficients can be obtained from suppliers of rotor brakes and will vary depending on specific liner/disc material combinations. Lining life and wear prediction calculations are considered proprietary by the lining manufacturers; therefore, they should be consulted in any application where guaranteed lining life or a minimum number of stops is specified. Helicopter rotor brake analysis differs from aircraft wheel brake analysis in two respects. First, advantage is taken of the aerodynamic drag on the rotating rotor blades, which varies with the square of the rotor rpm. This force alone would stop the rotor from full rpm in about 3 to 4 minutes. The second point to consider is that the turbine engines at idle produce a positive torque that varies inversely with rpm and which must also be reacted by the rotor brake.

CONNECTIONS

There are several different kinds of connections used in helicopter transmissions. These include bolts, studs, large diameter nuts, welds, and splines. Although all are not strictly related, this chapter will provide comments on the reliability aspects of each.

BOLTED CONNECTIONS

Bolted connections are used primarily to attach gear members to shaft flanges where one-piece construction is not possible or practical. The most common reliability problem associated with such connections is fretting under the bolt heads. At present there is no technique that can accurately predict the onset of fretting, nor are there any coatings available which will prevent fretting in all cases. It is known, however, that relative motion under under load between the attached surfaces causes fretting, and that the amount of motion necessary to produce fretting is very small, perhaps on the order of a few microns. There are some steps the designer can take that will minimize the possibility of fretting. First, if at all possible, the gearbox designer should design the connection so that all loads are reacted by friction between the clamped surfaces rather than by shear through the bolt body. A conservative approach should be used when analyzing the bolt circle for reacting the load in friction, since it is difficult to determine, particularly with radial loads, exactly how many bolts are reacting the load. With helical and spiral bevel gears, special effort should be made to be conservative with respect to preload, since there will be loads normal to the plane of the flange that will tend to separate the bolted surfaces. When titanium flanges are used, silver plate should be used under the bolt head, since titanium is much more susceptible to fretting than steel. Silver plate may also be used to minimize fretting on steel/steel connections. Webs and flanges that are offset, i.e., not centered under the gear, should be avoided if at all possible.

Care should be taken with bolted flanges to provide at least one full bolt diameter between the outer edge of the bolt hole and the edge of the flange. A lug analysis may be used to analyze this part of the connection, and this analysis should be performed even if the load is theoretically reacted by friction.

The bearing area under the bolt head should also be carefully checked for adequacy when analyzing bolted connections. This is especially true when softer materials such as magnesium are an element of the connection.

Studs and inserts are used to provide means of attachment to magnesium housings. Besides taking the same considerations as for bolted connections, the pull out strength of the studs and inserts must be considered. The literature provided by such manufacturers as Rosan^{R} contains the information necessary to select the proper stud or insert for any given application.

LARGE DIAMETER NUTS

Large diameter nuts are used for clamping bearings onto gear shafts and main rotor shafts and provide the means necessary to react thrust loads. The hand of the thread for large diameter nuts should be chosen so that in the event the bearing race turns on the shaft, the resulting motion will tend to tighten the nut on the shaft. A locking feature such as a small bolt through the nut or a tanged locking ring must be used. It is also advisable to use buttress threads for large diameter nuts. This type of thread produces very little radial load when torqued. This minimizes the reduction in shaft diameter which can adversely affect the press fit of adjacent bearings.

WELDS

Welding is used only to a very limited extent in helicopter transmissions. Although welded joints usually offer considerable weight savings over bolted connections and eliminate the possibility of fretting, the change in the metal structure in the weld zone makes accurate prediction of the strength of a welded joint difficult. This is probably why so few welds are found in current helicopter transmissions. Bolted connections are usually favored because there are coatings such as silver plating which can minimize fretting. When welds are used the design should be such that the exit side of the weld can be cleaned up. In other words, blind welds should be avoided.

INVOLUTE SPLINES

Involute splines are used to transfer torque between shafts and flanges, gears and shafts, and shaft and shafts. The design of splines is well covered by publications of such organizations as the American Gear Manufacturers Association and the Society of Automotive Engineerings. The most common problem associated with splines is wear due to fretting, particularly with loose splines such as those used on quill shafts. Attention to the following details can significantly lessen the chance of encountering spline problems. The most important consideration in the design of splines is strict observance of the limits on allowable bearing stress. With tight splines care should be taken to provide an adequate length pilot to react any type of bending loads. Avoid the use of pilots which are too short. With loose splines lubrication is a vital factor in determining how well the spline will perform. It is desirable, if possible, to keep the spline area flooded with oil at all times. Oil dams such as that shown in Figure 17 can be used to accomplish this. Crowning of loose splines is also necessary to prevent excessive wear. The amount of the crown should be carefully calculated on the basis of tolerances and full load deflections.

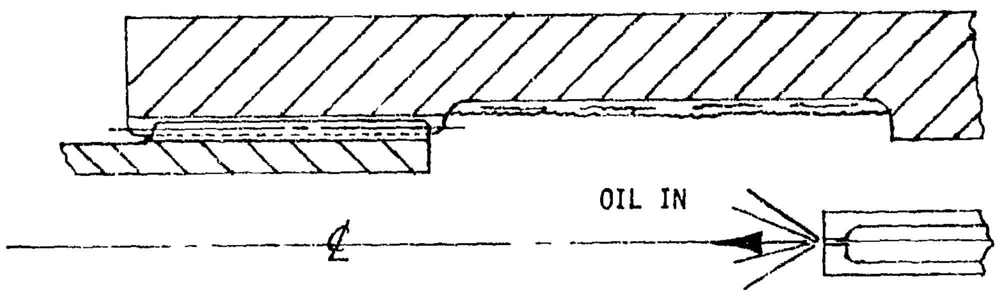


Figure 17. OIL DAM FOR LUBRICATION OF LOOSE SPLINES

SEALS

Seals in the past have been a continu. l reliability problem with helicopter gearboxes. Although seal leakage is not a serious failure mode, the amount of maintenance attributable to this problem has been a significant factor in the life-cycle maintenance costs of helicopter gearboxes. For this reason it is desirable to make field replacement of these seals as easy as possible. Further discussion of this is provided in the section on Maintainability. There are four basic seal types used on helicopter transmission gearboxes: conventional lip seals, hydrodynamic lip seals, circumferential seals, and face seals. These are illustrated in Figures 18 through 21. There are a number of factors that should be considered in the design of a seal for a specific application. These are discussed in the following paragraphs.

SPEED CAPABILITY

In seal selection, a primary consideration is the seal speed capability at the maximum expected transmission cavity pressure and shaft runout. It is stressed that both the pressure and runout must be considered, since they drastically affect the operation of both lip seals and circumferential seals. The speed capability of face seals is relatively insensitive to these two parameters, and it is this type of seal that is generally used for high-speed applications exceeding 15,000 ft/min. Conventional lip seals are generally limited to 3000 fpm. Hydrodynamic lip seals have a capability up to about 6000 fpm. Circumferential seals work satisfactorily up to 15,000 fpm.

PRESSURE CAPABILITY

Helicopter gearboxes generally have pressure diffentials between the cavity and outside, ranging from a fraction of a psi up to about 5 psi. Conventional and hydrodynamic lip seals have pressure capabilities that vary with speed. The higher the speed the lower the pressure capability. Circumferential seals have a very low pressure tolerance and should only be used where the pressure differential is less than 1 psi. Face seals can readily operate at all speed ranges at the highest pressures found in helicopter gearboxes. The transmission designer should pay particular attention to drainage in areas near external gearbox seals, since insufficient drainage is often the cause of high pressure differentials and subsequent seal leakage.

MISALIGNMENT AND SHAFT RUNOUT

Misalignment and shaft runout tend to disturb the lubricating film at the shaft seal interface. This leads to accelerated wear and increased leakage. Since the performance of all types of seals is affected by these factors, the worst case considering tolerances and stackups should be considered in the design of all seals.

Related to misalignmnet is the axial location of the seal with respect to the shaft. Again, careful checks should be made to insure that at the

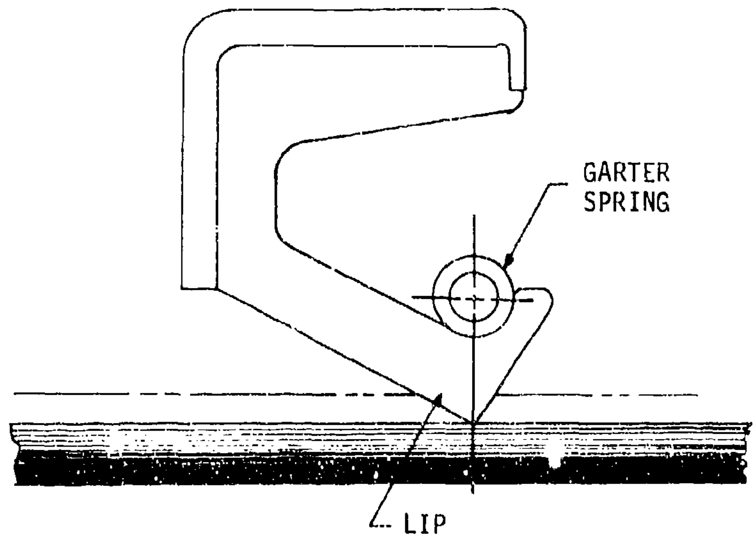


Figure 18. CONVENTIONAL LIP SEAL

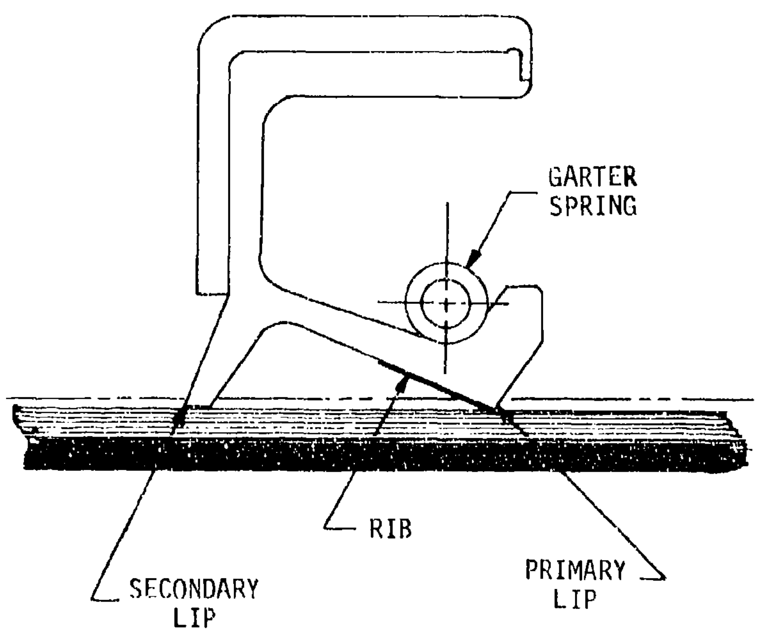


Figure 19. HYDRODYNAMIC LIP SEAL

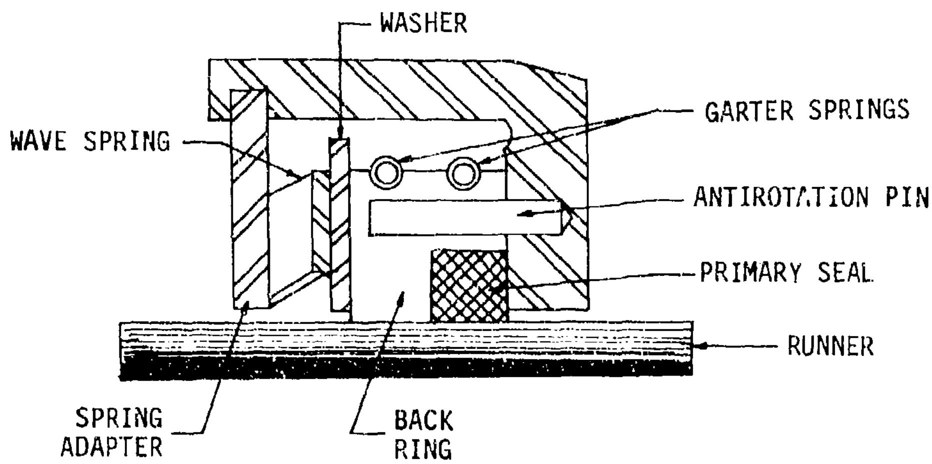


Figure 20. CIRCUMFERENTIAL SEAL

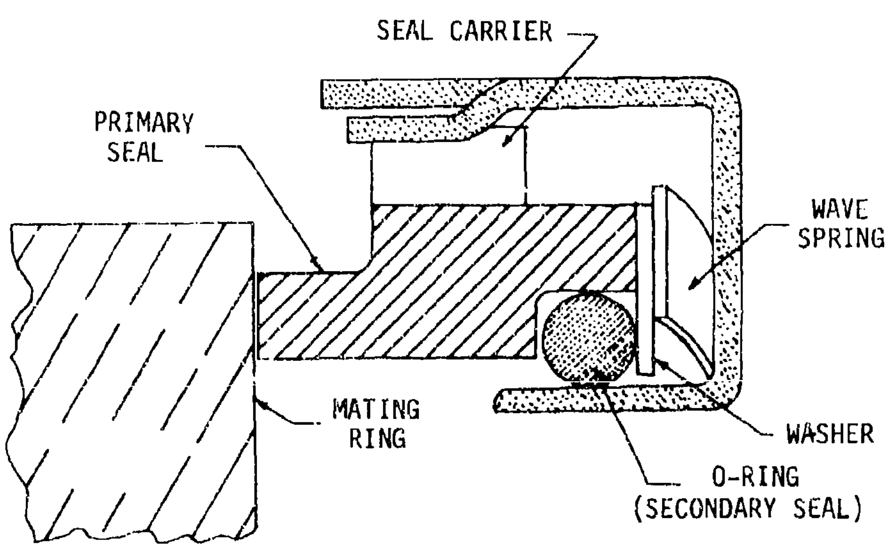


Figure 21. FACE SEAL

extreme positions of the seal relative to the shaft a suitable seal riding surface is present.

SEAL MATERIALS

There are three materials currently used for lip seals in helicopter transmissions: fluorelastomers such as Viton^R, silicone rubber, and nitrile. Fluorelastomers are currently used for most seals in helicopter transmissions. They have good resistance to most oils, fuels, and chemical solvents as well as to ozone, salt water and weathering. They are, however, costly and tend to be brittle at low temperatures. Silicone rubber is used to a very limited extent, since it has poor wear resistance and is only fair in its compatibility with most gearbox lubricants. Its only advantage is that it has excellent low temperature properties. This, however, is unimportant, since the lubricants become so viscous at low temperatures that leakage rarely occurs. Nitrile is rarely used in modern helicopter transmissions, although it is still employed on some older models. Prolonged exposure to air and oil at temperatures above 180^{o}F tend to harden this compound.

Some caution should be used when selecting seal materials. Various companies that manufacture oils to military specifications will use different additives even for oils of the same specification. In addition, the formulations for fluorelastomers vary from company to company. Hence, it is possible for one particular formulation of a given military specification lubricant to be compatible with a given seal material while another formulation of the very same lubricant could chemically attack the seal material. It is recommended that compatibility between seal materials and oil formulations, particularly with respect to additives, be carefully checked.

ENVIRONMENTAL CONSIDERATIONS

Exposure to various environmental conditions, such as water, sand and temperature extremes, is one of the direct causes of seal failure in helicopter gearboxes. It is imperative that secondary sealing devices be employed to screen the primary seal from contaminants that can lead to abrasion. Lip seals usually have secondary "dust lips" incorporated with the primary seal. Other exclusion devices that may be employed include sheet shrouds, wiper seals, slingers, labyrinths, and absorbent packings. The particular secondary seal chosen depends, of course, on the application. Such redundant seals are particularly important on main rotor shafts, where water seapage into the gearbox is a distinct possibility.

SACRIFICIAL RUNNERS

Seals are often designed to run directly on the shaft which they are sealing. This practice, however, can lead to the scrappage of the shaft if abrasive contaminants enter the seal and cause wear of the shaft itself. From a reliability viewpoint it is preferable to use thin sacrificial sleeves over shafts to provide a running surface for the seal. With this practice, only a relatively inexpensive sleeve, not an expensive shaft, need be replaced in the event of wear.

O-RINGS

O-rings are static seals used at housing interfaces. O-ring failure and attendant leakage are rarely caused by gearbox operation but by cutting of the O-rings during installation. Among the steps transmission designers can take to reduce the possibility of this happening are the following:

1. Don't use O-rings with a cross sectional diameter less than .10 inch.

2. Make certain that all edges that could possibly catch O-rings during installation are properly broken.

O-rings that are cut during installation can lead to potentially serious failures. These failures occur when pieces of O-ring contaminate the lubricant and lodge in lube jets, thereby blocking the flow of lubricant to dynamic components.

LUBRICATION SYSTEMS

The lubrication system of a helicopter drive system, shown schematically in Figure 22, serves two basic purposes. First, it provides the necessary lubricant film on the mating surfaces of dynamic components. Second, it provides the medium through which the heat generated by friction is removed from the gearbox to the outside air. It is importnat, from a reliability standpoint, that the lubrication system be properly designed, since an inadequate or poorly designed system can adversely affect the reliability of the gearbox.

SYSTEM DESIGN

A truly redundant lubrication system with two completely independent lubricant paths is very desirable but often not practical with the helicopter drive systems. More often, dual pumps are used, which feed into the same lubricant lines. If this is done, the pumps must be provided with one-way check valves which, in the event of failure of a single pump, prevent the working pump from pumping lubricant back through the failed pump, instead of into the lubrication system. A failure mode and effects analysis of the lubrication system can be a very useful tool in ensuring that the design of the pumps is such that this cannot happen. The pump should be protected from debris damage by a screen in the pump inlet line. This screen can be incorporated with a full-flow-type chip detector to save added parts. A shear area should be designed into the pump shaft so that in the event of pump seizure, the pump shaft shears, not some other component in the accessory drive train.

As fine a filteras possible should be incorporated into the lubrication system, because of the great impact filtration has on bearing life. Development is currently proceeding on filters as fine as 3 microns absolute. These should certainly be considered for all future gearbox designs.

SIZING AND POSITIONING OF JETS

The determination of how much lubricant is required for each dynamic component is determined by an efficiency or heat generation analysis. The oil flow required to remove the heat is calculated. Then, based on the specified pump pressure, the size of the jet can be easily found by using the equation for flow through an orifice. Care must be taken that the jet selected for any application is neither too large nor too small. A jet that is too small can lead to excessive heat generation in a bearing or scoring of gear teeth, while an oversized jet will lead to excessive churning losses. Jets with an orifice smaller than .040 inch should be avoided, since small pieces of debris can easily plug such small openings.

Primary power train gears should be lubricated both into and out of mesh, with perhaps 80 percent of the total for each mesh directed at the out-of mesh side, where cooling of the gear teeth is best effected. Rolling element bearings generally are first lubricated by jetting the required amount of oil into the gap between the cage and inner race. Some bearings, such as triplex and duplex bearings and overrunning clutch bearings, are

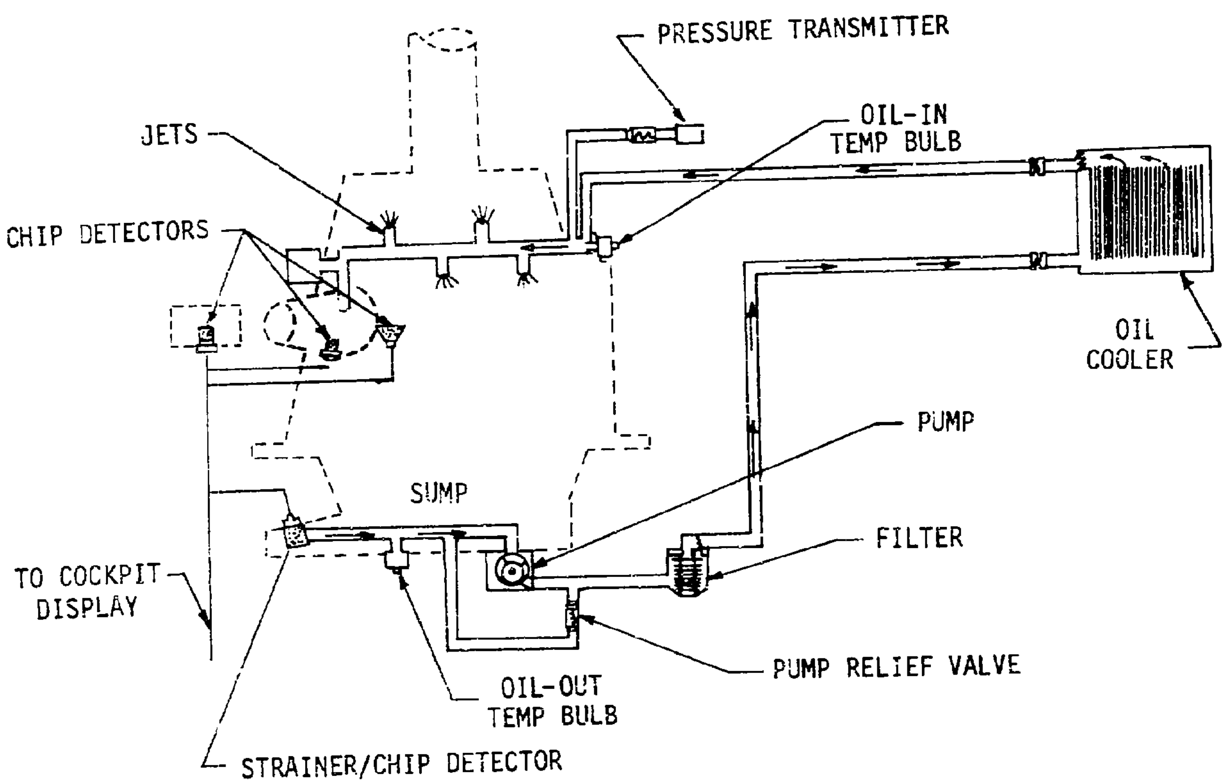


Figure 22. TYPICAL GEARBOX LUBRICATION SYSTEM

best lubricated through radial holes in the rotating shaft, as shown in Figure 23. This method, which relies on centrifugal force to provide positive lubrication, is very effective for hard-to-get-at bearings, and it does provide a good supply of lubricant to the critical inner race area. Planetary units are usually lubricated in a similar manner. With planetary units, it is customary to use a collector ring to create the required lubricant head.

WINDAGE AND CHURNING

Windage is nothing more than the air movement created by rotating machinery. Churning is the mechanical working of the lubricant, either by the rotating machinery or by the windage. Both windage and churning can drastically affect the performance of the gearbox. In order to avoid these phenomena, there are several steps the designer should take. First, he should make certain that sufficient drainage is provided at all cavities where lubricant may accumulate. Drain paths should be provided from both sides of rolling element bearings, even if oil is supplied at only one side. In areas where oil must drain past a high-speed gear, the drain passage should be sheltered from the windage. A large clearance between the gear and housing should be allowed at all high-speed gear meshes.

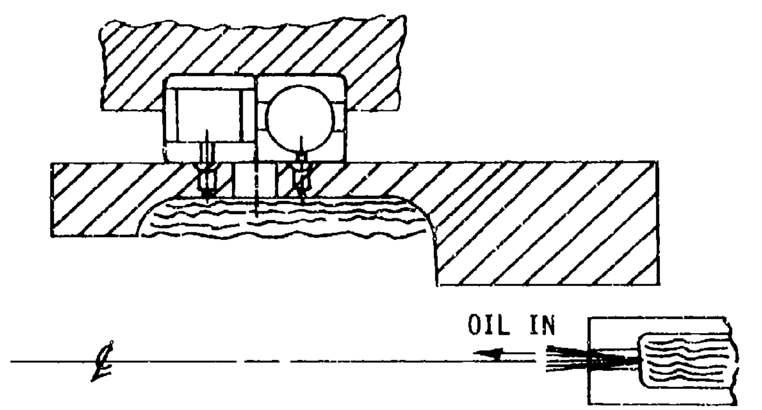


Figure 23. THROUGH-SHAFT BEARING LUBRICATION

MAINTAINABILITY

Designing maintainability into a helicopter drive system is a relatively easy task, if the program manager establishes a high priority for maintainability, and the gearbox designer maintains an awareness of this characteristic during the design stage. A little thought during design can save many hours of maintenance during the operational life of the gearbox.

ACCESSIBILITY

By far the most common problem noted in earlier drive system designs is the lack of accessibility provided for even routine maintenance tasks. Often such simple tasks as filter replacement, require maintenance personnel to be part contortionist because of obstruction by airframe, controls, or even other drive system components. Another common problem is the replacement of one defective component requiring the prior removal of other perfectly good components in order to effect the repair. Such problems as these can easily be avoided, if accessibility is a prime consideration during design.

Accessibility to drive system components on the aircraft is largely a matter of communication between the transmission designer, the aircraft configuration manager, and the designers of other systems such as the airframe. By ensuring that other systems do not interfere with routine maintenance and inspection of the transmission system and vice versa, many field problems can be avoided. The more awkward or difficult a maintenance task is to perform, the more likely it will be performed incorrectly or not at all. A full size mock-up can be very useful in pinpointing potential accessibility problems.

Accessibility to the following components is especially important.

- Oil filter
- Oil filler cap and sight windows (Note: These should be located so that a person adding oil to the gearbox can easily see the sight window).
- Gearbox mounting hardware
- Chip detectors
- Tail drive shaft hanger bearings
- Oil cooler air inlet (so that it can be easily cleared of debris)
- Accessories such as oil pump, hydraulic pumps, and generators

MODULARIZATION AND FIELD REPLACEABLE ITEMS

Modularization is simply designing the main gearbox so that such sections as inputs, accessories and tail takeoff may be removed from the main part of the gearbox as self-contained modules. Figures 24 and 25 illustrate this concept. Note that quill shafts splined on either end provide the necessary torque connections between the modules. Screens are provided at each module interface so that a failure within a single module will not contaminate the rest of the gearbox. In addition, each module is provided with its own chip detector so that a failure can easily be isolated to a particular module. These modules should be designed so that they may be removed using only standard wrenches found in an aircraft repairman's tool box. Care must be taken that lubrication transfer passages between the modules are well sealed.

Modularization is an important concept that can lead to substantial savings in maintenance costs by substantially reducing aircraft downtime. It should be seriously considered for all but the very low power helicopter drive systems.

While on-aircraft repairs are not practical for the majority of helicopter drive train components, there are a number of items that should be easily field replaceable. Some of these, such as tail drive shafts, couplings and hanger bearings, and oil coolers, have always been field replaceable and should remain so. Careful attention should be paid to the design of such items to facilitate their replacement. There are two other components, which, in the past, have not usually been field replaceable, but perhaps should be. These are freewheel units and external seals.

Freewheel units, because of their relatively low primary failure rate, were not detrimental to the overall maintainability of the drive system. However, new generation gearboxes will be exclusively on-condition maintenance items. This policy will lead to time accumulation on some gearboxes far beyond that seen in the past; hence, wear-induced failure modes will become more prominent. The problem here is that all types of freewheel units, whether they be sprag, ramp roller, or spring, experience wear during the overrunning mode. With the higher times accumulated on on-condition gearboxes, it is more likely that wear will progress to the point where the freewheel unit will not operate satisfactorily. In other words, it is likely that excessive freewheel unit wear will become more prominent as a primary failure mode, thus making field replaceability of these items a cost-effective design feature.

External seals of helicopter gearboxes have not in the past usually been designed as field replaceable items. Often however, to avoid costly overhauls simply because of seal leakage, the replacement of these seals has been assigned to the organization maintenance level. This has led to numerous cases of incorrect installation, damage during installation, or gearbox contamination during seal replacement. A more desirable practice, from the maintainability point of view, would be to design seals premounted in a small housing, which pilots on the main housing and bolts in place. Replacement is accomplished by simply removing the bolts attaching the seal/housing assembly, sliding out this assembly, and replacing it with a new

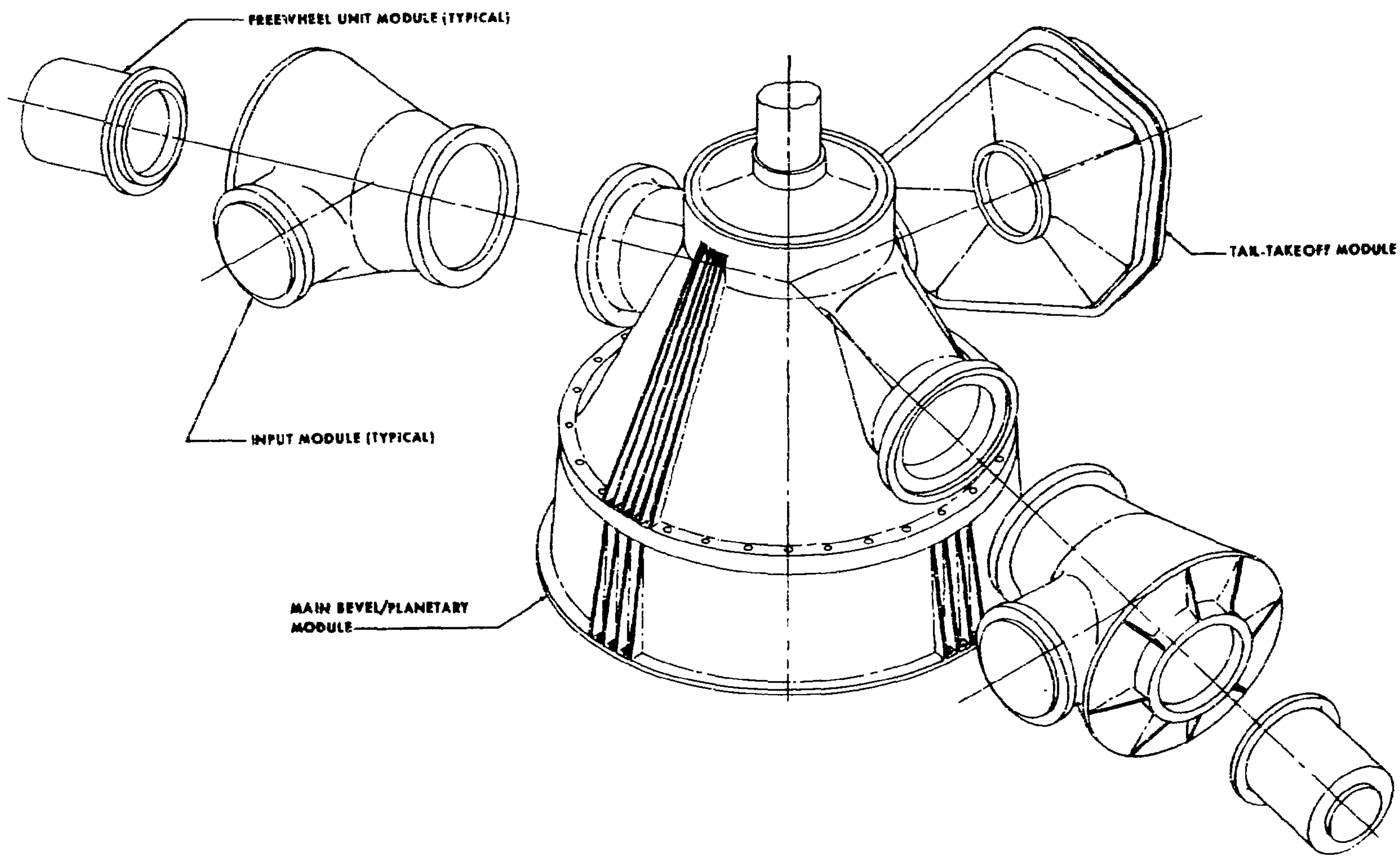


Figure 24. CH-54 MODULARIZED GEARBOX ISOMETRIC

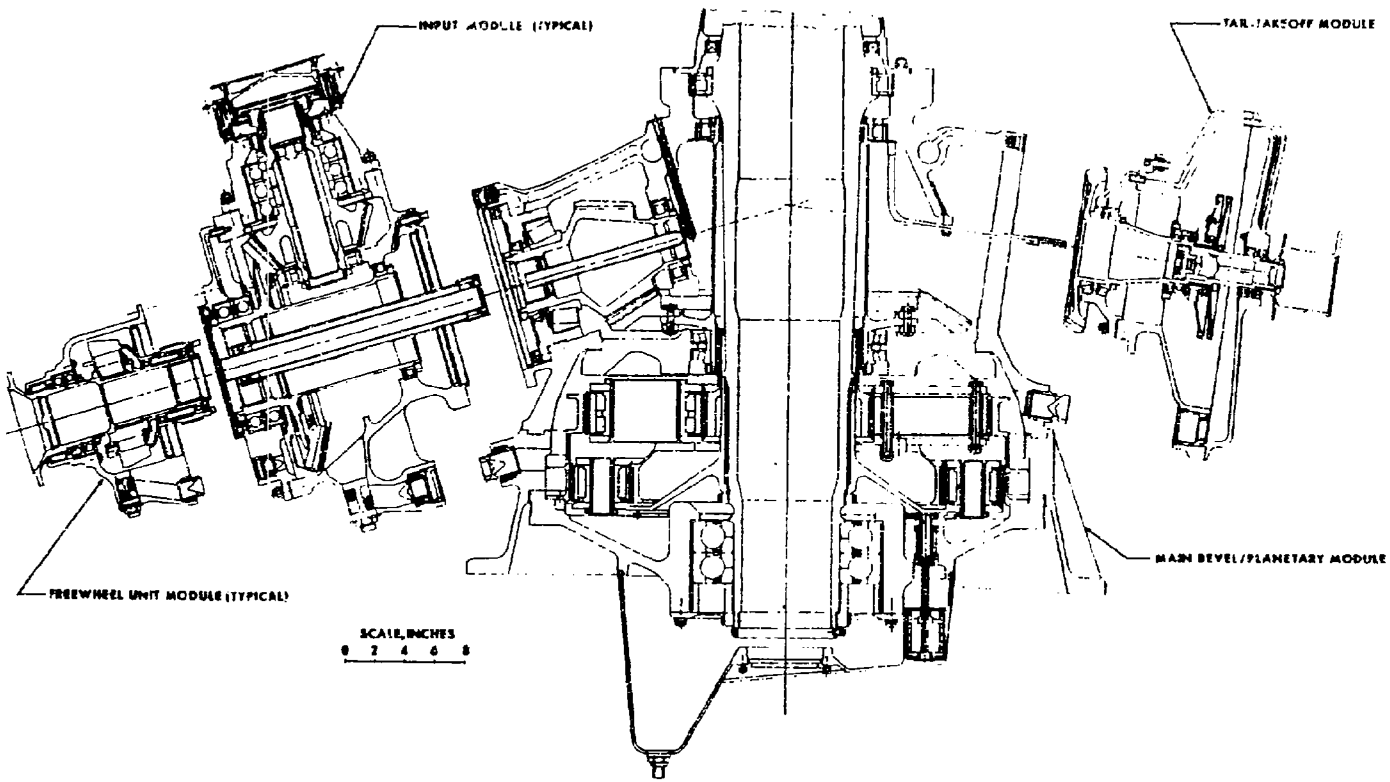


Figure 25. CH-54 MODULARIZED GEARBOX CROSS SECTION

seal/housing assembly. This assures proper alignment of the seal with the shaft centerline and minimizes handling of a seal surface by the aircraft mechanic.

INTERCHANGEABILITY

Interchangeability is another way to simply and conveniently reduce maintenance costs. The use of interchangeable parts can significantly reduce spares requirements as well as considerably simplify maintenance. Among the items that lend themselves to interchangeability are tail drive shaft sections, drive shaft flanges, large diameter nuts, spacers, input and output intermediate gearbox housings, and all input components of multi-engine helicopters. Using identical bearings in as many places as possible also promotes the interchangeability concept. Bearings have become extremely long lead-time items and maintaining adequate supplies of many different bearings has become a serious problem at the depot level. Using the same bearing in as many applications as possible will not only reduce the number of different bearings that must be stocked, but will also permit the ordering of larger quantities of bearings with a substantial reduction in the price per bearing. This same practice can also be effectively utilized with seals.

STANDARDIZATION

Standardization is yet another concept that can considerably improve the maintainability characteristics of a drive system design. It can simplify many maintenance tasks as well as reduce the time to complete them. Like interchangeability, standardization can also have a beneficial effect in the supply of replacement parts. Standardization should be employed to the maximum extent possible with such components as nuts, bolts, and studs. This will reduce the number of special tools required for drive system maintenance. Some earlier Army aircraft drive systems were particularly bad with respect to lack of standardization of hardware. They required the aircraft mechanic to be equipped with many special tools, which often added more than 100 pounds to the weight of his tool kit. Such situations could have been avoided, if the designers of these drive systems had maximized standardization during the design.

OVERHAUL ENHANCEMENT DESIGN FEATURES

Attention to the above practices can considerably simplify the task of overhauling helicopter gearboxes. There are, however, some special considerations that are especially pertinent to facilitating gearbox overhaul. The first of these is rather simple and should be adopted as a general design practice.

Flats should be provided on shaft shoulders so that a bearing puller can easily remove the bearing without damaging it. If this is not done, the bearing puller will often slip off the inner race and damage the cage, thereby ruining an otherwise perfectly good bearing. As was mentioned earlier, bearings have become very expensive, long lead-time items. The damaging of bearings due to inattention to removal procedures during design

is not excusable.

Another practice that is especially costly and time consuming at overhaul is the requirement that magnesium housings be completely stripped of their protective coatings prior to inspection. It is extremely unlikely that the housing could be cracked without a corresponding crack in the protective paint. If certain areas of the housing had been touched up because of corrosion, these areas alone could be stripped. Magnetic particle inspection of the housing could then be performed. If the housing is not cracked, which is virtually always the case, the housing could be returned to service with relatively little work. Thus magnetic particle inspection over the paint could save two very expensive and time-consuming steps: stripping before inspection and reapplying coatings after inspection.

Container design, although not strictly the responsibility of the transmission designer, is another area where the improvement could result in considerable cost savings at depot. Housing corrosion is generally only a minor problem at the organizational level. There are occasional small areas of corrosion but severe corrosion is relatively rare. Virtually all gearboxes arriving at depot for overhaul, however, are found to have corrosion, and severe corrosion is not uncommon. The reason for the apparent discrepancy between corrosion at the two maintenance levels seems to be the ineffectiveness of the gearbox containers in keeping out moisture. This is supported by the fact that accumulations of water are often found in the bottoms of the containers when they are opened at depot.

Another practice that can substantially reduce depot maintenance costs is the provision for extra material in those areas where fretting, corrosion, or wear may be expected. By providing extra material in such areas the part can often be reworked and reused without sacrificing its structural integrity. This can lead to considerable savings in replacement parts at depot.

Somewhat related to the above is the practice of using the gearshaft as the inner race of the bearing. Although this practice is desirable from the weight standpoint since it eliminates the inner race, nut, and locking device and usually results in a shorter shaft, it is not cost effective from the overhaul standpoint. A spall on the bearing race surface of the gearshaft results in the scrapping of a very expensive gear and bearing instead of just the bearing alone. The practice of using integral bearing races on shafts could, however, be cost effective if a satisfactory procedure were developed where the shaft could be reworked and the bearing replaced by one with oversize rolling elements.

One final point should be made with respect to overhauling of helicopter gearboxes. In most cases product support personnel, not gearbox designers, set up overhaul procedures. While these people may be very competent and well experienced in the preparation of such documents, the gearbox designer is certainly more qualified to say how his gearbox should be overhauled. Hence, it is recommended that overhaul procedures be prepared by the gearbox design engineers, or if that is not feasible, at least be carefully reviewed and approved by them.

DIAGNOSTICS

Diagnostic techniques for use in helicopter gearboxes can be divided into two basic groups: debris analysis and vibration analysis. The debris analysis methods include chip detectors, filter checks, and SOAP (Spectrometric Oil Analysis Program). Vibration analysis consists of the monitoring of certain frequency bands where changes in amplitude indicate failures or impending failures. In addition to these techniques, the standard lubrication system pressure and temperature sensors are employed.

CHIP DETECTORS

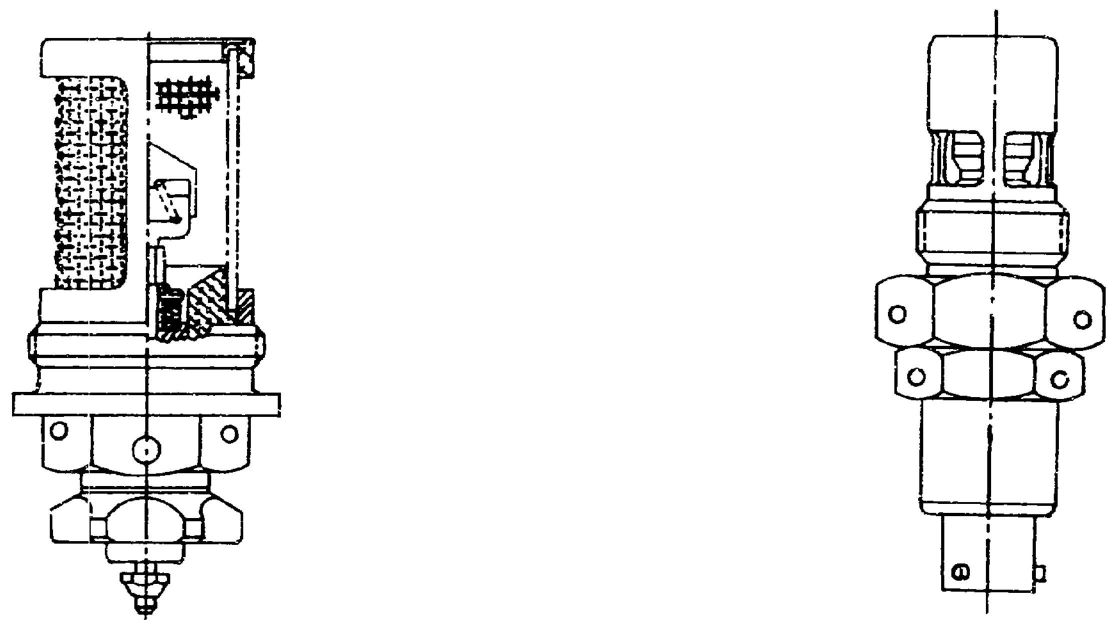
By far the most commonly used failure detection device in helicopter gearboxes is the magnetic chip detector. In its simplest form the chip detector is a magnetic plug that collects ferrous debris on a powerful two-pole magnet. This type of chip detector must be removed and visually inspected to determine component condition. Current transmissions use electric chip detectors that are remotely monitored and checked periodically by a continuity test. These chip detectors can be procured with a variety of features such as a self-closing feature for quick removal without lubricant drainage, and the inclusion of a high-temperature warning switch.

The two types currently used in most present transmissions are the full-flow and the sump-mounted plug shown in Figure 26. The full-flow chip detector is usually installed in a lubrication line leading to the pump and monitors all the lubricant that is continously circulating in the transmission. A strainer incorporated with the chip detector retains large particles, thus eliminating the need for a separate pump inlet screen. The sump-mounted chip detector magnetically attracts particles that settle to the bottom of the sump. Its efficiency is less than the full-flow chip detector, since it depends mostly on magnetic attraction to capture debris particles, while the full-flow chip detector has the advantage of pump pressure carrying the debris particles to it.

Chip detector sensitivity depends on the gap width, gap area, the type of magnetic circuit, and the shape of the electrodes. In general, smaller gap widths and areas are more sensitive, as are closed (vs. open) magnetic circuits and sharp (vs. rounded) electrodes.

Recently, chip detection systems have been developed that are capable of generating capacitor discharge pulses after the chip detector gap has been bridged by conducting particles. If this conductive path consists of "fuzz" or fine slivers, it will be interrupted by the current pulse.

Two types of systems can be employed with capacitive discharge chip detectors: manual and automatic. In the manual system, whenever the gap is bridged the chip detector light is activated. The pilot then activates the discharge mechanism. If the gap has been bridged by fuzz, it will be burned up and the light will go out. In the automatic system, the discharge mechanism is automatically activated, whenever the gap is bridged. The chip light will not go on unless the discharge fails to interrupt the conductive path.



FULL-FLOW SCREENED

PLUG TYPE

Figure 26. TYPICAL CHIP DETECTORS

Early experience with capacitive discharge chip detectors has been very encouraging. Several hundred hours of accumulated bench and flight testing have not produced a single false indication, the problem which has long plagued conventional chip detectors.

FILTER CHECKS

Filter checks have been recently employed during the development of superfine oil filters as a supplement to chip detectors. With this technique a relatively coarse screen (about 80 microns) is placed around the superfine filter element. This screen, which traps only large particles, is inspected periodically and after chip light activation to aid in the determination of whether or not a failure has indeed occurred. The inspection of the debris is done visually with a simple means of magnification to determine the types of debris trapped by the mesh.

SPECTROMETIC OIL ANALYSIS PROGRAM (SOAP)

Another detection method, used to a much lesser extent than chip detectors, is the spectrometric oil analysis program commonly known as SOAP. The SOAP technique consists of performing a spectrometic analysis of a gearbox oil sample to determine metallic content. The SOAP sample is taken from a subject gearbox as soon as possible after a flight to prevent metallic particles from settling out of the lubricant. A portion of the sample lubricant and an oil standard with known metallic content are placed on a film plate to record the wave lengths of the different elements. The exposed film is then placed in an optical comparator, which permits comparison of the widths of the lines representing the various elements. This enables the technician to determine the contaminant levels in the SOAP sample. If the SOAP technique is to be used on a given gearbox, the transmission designer should provide means to facilitate the taking of the oil sample. With some past designs the taking of the sample is extremely awkward, and this can affect the results of the SOAP analysis.

The SOAP technique has been used to a limited extent by commercial operators, the Army (ASOAP) and the Navy (NOAP). The success has been somewhat limited. There appears to be two basic problems with SOAP. First, the proper threshold level for gearbox removal has been difficult to establish. It appears that although some gearboxes exhibit high iron content levels, the gearbox turns out to be perfectly acceptable when it is disassembled and inspected. Other gearboxes have exhibited fretting failures, although the iron content has been relatively low. The Army, in particular, because of its large number of flight facilities, has experienced logistics problems with its ASOAP program. This is due to communication problems between the ASOAP laboratory and the various operational activities, and has sometimes led to needless gearbox removal.

A more serious problem, perhaps a more pertinent one as far as the future of SOAP is concerned, is the advent of superfine oil filters with the capability of removing all particles greater than 3 microns from the lubrication system. Although such filters are currently only in the development stage in helicopter gearboxes, it seems certain that they will become

standard equipment within a few years because of the great impact they have in improving bearing life. The problem this raises with SOAP is that the technique cannot be used to analyze such fine particles; hence, the large scale incorporation of superfine filters will be tantamount to ending the practicality of SOAP.

Oil Debris Monitoring (Particle Count)

This technique, like SOAP, is not an in-flight monitoring technique. It consists essentially of determining the distribution of particle sizes within the gearbox lubricant. The primary problem with this method is that the relationship between particle size distribution and failure modes is largely unknown. Hence, this method may result in a large number of false removals and/or missed failures because the threshold for gearbox removal is mostly guess work. This may be a moot point, however, since superfine filters would also render this method impractical.

Vibration Analysis

The development of failure detection by vibration analysis has been the major thrust of a number of programs to develop an AIDAPS (Automatic Inspection, Diagnosis, and Prognosis System) for helicopter drive systems. Various techniques including low-frequency analysis, high-frequency analysis, narrow band spectrum analysis, and shock pulse monitoring have been evaluated and some of the more promising techniques have been tested in an actual helicopter transmission. Testing to date has been largely conducted with discrepant components implanted in the gearbox in order to define differences in vibration signatures between assemblies with defective parts and those without. There has been no extensive study on a fleet of helicopters that would provide an indication of the success rate of such a system.

The instrumentation and electronics hardware required for an AIDAPS are of necessity very sophisticated, a fact which could jeopardize the reliability of the system should it ever be widely employed. Another problem found with the application of AIDAPS to helicopter transmissions is the fact that vibration signatures of helicopter gearboxes are very maintenance sensitive and vary considerably from aircraft to aircraft, even among like models. Hence, in order to incorporate such a vibration detection system, a large amount of data would have to be assembled to establish vibration limits for gearbox removal. Thus, not only would the hardware itself be expensive (probably greater than $10,000 per aircraft), but also the research and development costs required to develop an AIDAPS so it could be practically employed on production aircraft would probably be prohibitive.

Evaluation

From the above discussion, a number of conclusions can be drawn regarding the best policy to adopt with respect to drive system diagnostic systems. First, it is not recommended that investigations of SOAP or particle count techniques be pursued to a greater extent than they now are. In all likeli-

hood, both of these techniques will soon be rendered obsolete by the incorporation of superfine filters. Second, before committing any large amounts of resources to the development of AIDAPS, time should be allowed to evaluate how effective fuzz burn-off chip detectors are in reducing false indications on the UH-60A BLACK HAWK. There is no sense in spending large amounts of R&D funds developing a new expensive complicated system when it is [illegible]y that a much cheaper and simpler existing system will do the job just as well, if not better. Some may point out that chip detectors only provide after the fact indication of failures. However, given the fact that little is known about the symptom/failure relationship before failures occur, it is unlikely that any system no matter how sophisticated could give reliable advance warning of failure in the foreseeable future. Another justification often cited for supporting the development of AIDAPS is that chip detectors do not point out exactly which component failed within the gearbox. For helicopter gearboxes, however, this information is of academic interest only. Any failure which occurs inside a helicopter gearbox requires the removal of either the entire gearbox or the entire module in a modularized gearbox. Since the defective gearbox or module will be completely disassembled and inspected at depot anyway, there is no cost saving in knowing at the time of failure exactly which component failed. This type of information is useful only for failures which can be repaired on the aircraft, of which there are very few in helicopter gearboxes.

The most promising approach to diagnostics at this time appears to be a system which incorporates both sump mounted and flow through fuzz burn-off chip detectors supplemented by filter inspection. This system has many advantages over the other candidates. First, it is simple and inexpensive. Second, it is a proven system with respect to flight safety. Third, if early experience with fuzz burn-off chip detectors is any indication, it will be very reliable and produce very few unnecessary gearbox removals.

HAZARD FUNCTION ANALYSIS

This chapter defines a step-by-step procedure for quantitatively evaluating the reliability of a gearbox during the design stage. This procedure will relate the reliability of a new design to the reliability of an actual gearbox of similar design with a known MTBR (Mean Time Between Removal) by determining a correlation factor, K. The technique may also be used to predict the probability of a gearbox passing a test of a given length and power spectrum.

The reliability is calculated for each failure mode anticipated, such as gear wear, bearing spalling, seal leakage, spline wear, housing corrosion, etc., using the Weibull reliability function given by Equation (1).

$$R(T) = e^{-\left[\frac{T}{\theta}\right]^{\beta}} \qquad (1)$$

where $R(T)$ = reliability at T hours

θ = size parameter

β = shape factor

Recommended size parameters and shape factors for each failure mode, together with observed high and low limits, are given in Table 1. Except for bearing spalling, these values are based on historical experience from previous helicopter drive systems using a hazard function analysis[1]. The size parameter, θ, for bearing spalling is based on the adjusted L_{10} bearing life including material factor, processing factor, EHD factor, etc. The value of the shape factor, β, for bearing spalling is 10/9. This value is commonly accepted within the bearing industry and is believed to be somewhat conservative, since values ranging from 1.3 to 1.6 have been measured in service.

The first step in the reliability prediction procedure is to determine the correlation factor. This is done by performing the analysis on an existing gearbox with a known MTBR as follows using Steps 1-5. If a gearbox of similar design does not exist or the MTBR of the similar gearbox is unknown, the reliability of the new design can still be estimated by assuming the correlation factor is equal to 1.0 and proceeding with calculations using Steps 6-9.

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[1] Trustee, B., HELICOPTER DRIVE SYSTEM ON-CONDITION MAINTENANCE CAPABILITY, Sikorsky Aircraft Division, United Technologies Corporation; USAAMRDL Technical Report 75-57, Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, July 1976, AD A028414.

TABLE 1. SIZE AND SHAPE PARAMETERS FOR TRANSMISSION FAILURE MODES

FAILURE MODE	Recommended		High		Low	
	θ	β	θ	β	θ	β
Spur Gears						
Excess Wear	5.7×10^5	1.3	9.6×10^5	1.3	1.5×10^4	1.3
Other Modes (Broken Tooth, Pitting, etc.)	1.3×10^8	1.0	2.5×10^8	1.0	4.3×10^5	1.0
Spiral Bevel Gears						
Excess Wear	8.4×10^5	1.4	2.1×10^6	1.4	3.2×10^5	1.4
Other Modes (Broken Tooth, Pitting, etc.)	1.3×10^6	1.0	4.6×10^6	1.0	1.3×10^4	1.2
Ball Bearings						
Spalling	$7.58 \times L_{10}$	10/9	$7.7 \times L_{10}$	1.1	$4.1 \times L_{10}$	1.1
Other Modes (Broken Cage, Smearing, etc.)	3.2×10^4	1.3	2.8×10^7	1.0	1.1×10^4	1.7
Roller Bearings						
Spalling	$7.58 \times L_{10}$	10/9	$7.7 \times L_{10}$	1.1	$4.1 \times L_{10}$	1.1
Other Modes (Broken Cage, Smearing, etc.)	1.6×10^6	1.1	8.2×10^8	1.0	5.3×10^4	1.1
Tapered Roller Bearings						
Spalling	$7.58 \times L_{10}$	10/9	$7.7 \times L_{10}$	1.1	$4.1 \times L_{10}$	1.1
Other Modes (Broken Cage, Smearing, etc.)	2.5×10^6	1.0	2.0×10^7	1.0	3.8×10^5	.7
Seals						
Lip Seal Leakage	1.7×10^6	.5	4.0×10^7	.4	1.4×10^5	.7
Face Seal Leakage	1.0×10^4	1.5	1.1×10^6	1.1	2.7×10^3	2.0
O-Ring Leakage	8.8×10^7	.74				
Shaft Assembly						
Shaft Crack/Fracture	2.0×10^7	1.0	4.9×10^7	1.0	4.8×10^3	1.0
Spline (Loose) Fretting/Wear	1.8×10^4	1.7	6.0×10^6	1.2	1.7×10^3	3.1
Flange Crack/Fracture	8.8×10^5	1.0				

TABLE 1. (Continued)

FAILURE MODE	Recommended		High		Low	
	θ	β	θ	β	θ	β
Housing Assembly						
Housing Crack/Fracture	9.8×10^4	1.7	9.1×10^5	1.6	2.8×10^3	2.2
Housing Corrosion	1.5×10^4	1.3				
Bearing Retention Clip Fracture	6.1×10^4	1.3	2.8×10^7	1.0	3.2×10^4	1.6
Ramp Roller Clutch						
Excess Wear (Roller, Cam Shaft)	6.4×10^3	1.5	3.8×10^7	1.1	2.4×10^3	2.6
Cage Fracture	4.8×10^3	1.6	1.3×10^5	1.8	3.9×10^2	1.8
Other Modes	2.1×10^4	1.5	3.0×10^6	1.1	7.6×10^3	1.6
Sprag Clutch						
Excess Wear (Sprag, Shaft)	6.4×10^3	1.5	3.8×10^7	1.1	2.4×10^3	2.6
Other Modes	4.5×10^3	1.6	2.1×10^5	1.5	3.9×10^2	1.6
Planetary Assembly						
Cage Plate Crack/Fracture	1.2×10^6	1.0				
Thrust Washer Wear	1.8×10^4	2.7				
Retaining Ring Fracture	5.2×10^5	1.0	2.5×10^7	1.0	2.0×10^4	1.0
Lubrication System						
Oil Pump Low Pressure	2.5×10^7	.63				
Oil Jet Plugged	3.5×10^5	.64	1.6×10^9	.6	4.4×10^4	.6

STEP 1

Compute the theoretical reliability of a gearbox with a known MTBR at 100 hours using the following equation:

$$R(100) = e^{-\Sigma\left[\frac{100}{\theta_i}\right]^{\beta_i}} \qquad (2)$$

where θ_i = size parameter associated with i^{th} failure mode (tabulated in Table 1)

β_i = shape parameter associated with i^{th} failure mode (tabulated in Table 1)

The summation in Equation (2) consists of all possible failure modes in the gearbox.

STEP 2

Similarly, compute the theoretical reliability of the gearbox with the known MTBR at 10,000 hours using the following equation:

$$R(10,000) = e^{-\Sigma\left[\frac{10,000}{\theta_i}\right]^{\beta_i}} \qquad (3)$$

STEP 3

Compute the composite shape and size parameters, β and θ , for the gearbox with the known MTBR from the calculated values of reliability at 100 and 10,000 hours using the following equations:

$$\beta = \frac{\ln\ln\frac{1}{R(10,000)} - \ln\ln\frac{1}{R(100)}}{4.605} \qquad (4)$$

$$\theta = e^{\frac{4.605\beta - \ln\ln\frac{1}{R(100)}}{\beta}} \qquad (5)$$

STEP 4

Compute the theoretical MTBR of the gearbox with the known MTBR from the shape and size parameters calculated in Step 3 using the following equation:

$$MTBR = \Theta\cdot\Gamma(1 + 1/\beta) \qquad (6)$$

where $\Gamma(1 + 1/\beta)$ = the gamma function of the quantity $(1 + 1/\beta)$ which can be found using Figure 27.

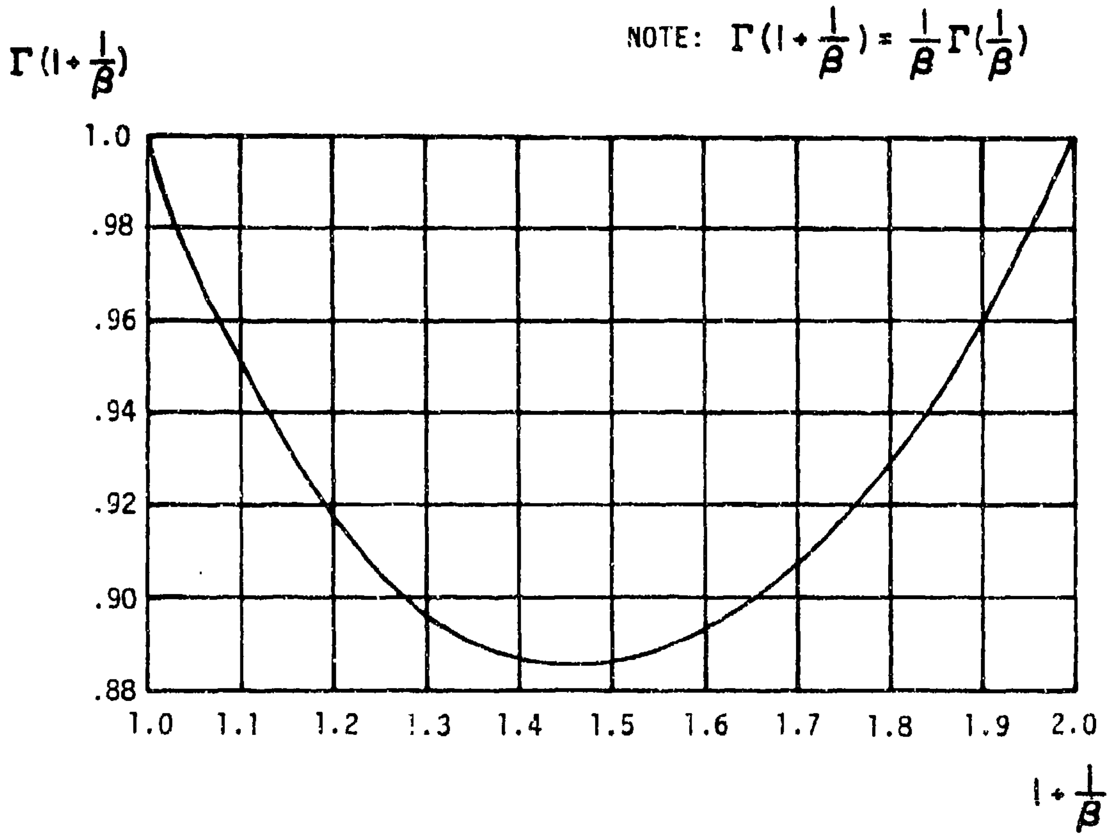

Figure 27. GAMMA FUNCTION

STEP 5

If the gearbox with the known MTBR was maintained on-condition a correlation factor K may be simply found by the following:

$$K = \left[\frac{MTBR_{observed}}{MTBR_{theoretical}}\right]^{\beta} \qquad (7)$$

If the gearbox with the known MTBR has a fixed TBO (Time Between Overhaul), then an adjusted MTBUR (Mean Time Between Unscheduled Removals) must be used to determine the correlation factor K. This can be found from the following expression using the value of β from Step 3 and the gamma function from Step 4.

$$MTBUR_{adjusted} = \left[MTBUR_{observed} TBO^{\beta-1}\right]^{1/\beta} \Gamma(1 + 1/\beta) \qquad (8)$$

The correlation factor K can then be found from the following:

$$K = \left[\frac{MTBUR_{adjusted}}{MTBR_{theoretical}}\right]^{\beta} \qquad (9)$$

STEP 6

Compute the reliability of the new design at 100 hours with the following equation:

$$R(100) = e^{-1/K \; \Sigma\left[\frac{100}{\theta_i}\right]^{\beta_i}} \qquad (10)$$

The calculation is done using the same basic method as that used in Step 1.

STEP 7

Similarly, compute the reliability of the new design at 10,000 hours with the following equation:

$$R(10{,}000) = e^{-1/K \; \Sigma\left[\frac{10{,}000}{\theta_i}\right]^{\beta_i}} \qquad (11)$$

STEP 8

Compute the composite shape and size parameters, β and θ, of the new design from the calculated values of reliability obtained in Steps 6 and 7 using Equations (4) and (5).

STEP 9

Compute the MTBR of the new design from the calculated values of β and Θ obtained in Step 8 using Equation (6).

Example - Reliability Prediction of New Intermediate Gearbox

To illustrate the reliability prediction procedure, the MTBR of the newly designed intermediate gearbox will be determined. The correlation factor for a similar gearbox with a known MTBUR and TBO must be obtained before the MTBR of the newly designed gearbox can be determined. A parts list and corresponding values for size and shape parameters are given in Table 2 for the gearbox of known MTBR shown in Figure 28. Similar data is given in Table 3 for the new gearbox shown in Figure 29. Experience shows that the existing similar design has an observed MTBUR of 4000 hours and a TBO of 2000 hours. The calculations, starting with Step 1, are outlined below.

STEP 1

Compute the theoretical reliability of the existing gearbox at 100 hours using Equation (2). Values of the exponent and the reliability for each failure mode are given in Table 2.

$$\Sigma(100/\Theta_i)^{\beta_i} = .02110 \qquad R(100) = e^{-.02110} = .9791$$

STEP 2

Compute the theoretical reliability of the existing gearbox at 10,000 hours using Equation (3). Values of the exponent and the reliability for each failure mode are given in Table 2.

$$\Sigma(10{,}000/\Theta_i)^{\beta_i} = 2.1394 \qquad R(10{,}000) = e^{-2.1394} = .1177$$

STEP 3

Compute the composite shape and size parameters, β and Θ , for the existing gearbox using the values calculated in Steps 1 and 2 by substitution in Equations (4) and (5).

$$\beta = \frac{\ln\ln\frac{1}{.1177} - \ln\ln\frac{1}{.9791}}{4.605} = 1.003$$

$$\Theta = e^{\frac{4.605(1.003) - \ln\ln\frac{1}{.9791}}{1.003}} = 4679$$

TABLE 2. RELIABILITY ANALYSIS FOR EXISTING GEARBOX

PART	i	QUANTITY (N)	FAILURE MODES	θ_i	β_i	$N(100/\theta_i)^{\beta_i}$	$N(10000/\theta_i)^{\beta_i}$	$R_i(100)$	$R_i(10000)$
Spiral Bevel Gears	1	2	Excess Wear	8.4×10^5	1.4	6.41×10^{-6}	4.05×10^{-3}	.999994	.995962
	2	2	Other Modes	1.3×10^6	1.0	1.54×10^{-4}	1.54×10^{-2}	.999846	.984733
Tapered Roller Bearings			Spalling						
	3	1	15,830 hours	1.2×10^5	10/9	3.79×10^{-4}	6.32×10^{-2}	.999621	.938729
	4	1	804,750 hours	6.1×10^6	10/9	4.82×10^{-6}	8.04×10^{-6}	.999995	.999196
	5	1	39,580 hours	3.0×10^5	10/9	1.37×10^{-4}	2.28×10^{-2}	.999863	.977416
	6	1	606,860 hours	4.6×10^6	10/9	6.59×10^{-6}	1.10×10^{-3}	.999993	.998901
	7	4	Other Modes	2.5×10^6	1.0	1.60×10^{-4}	1.60×10^{-2}	.999840	.984127
Lip Seals	8	2	Leakage	1.7×10^6	.5	1.53×10^{-2}	1.53×10^{-1}	.984778	.857793
O-Rings	9	5	Leakage	8.8×10^7	.74	2.00×10^{-4}	6.03×10^{-3}	.999800	.993992
Shaft	10	2	Crack/Fracture	2.0×10^7	1.0	1.00×10^{-5}	1.00×10^{-3}	.999990	.999000
Flange	11	2	Crack/Fracture	8.8×10^5	1.0	2.27×10^{-4}	2.27×10^{-2}	.999773	.977529
Housings	12	3	Crack/Fracture	9.8×10^4	1.7	2.47×10^{-5}	6.19×10^{-2}	.999975	.939930
	13	3	Corrosion	1.5×10^4	1.3	4.45×10^{-3}	1.77	.995561	.170174
				TOTAL (Σ)		.02110	2.1394		

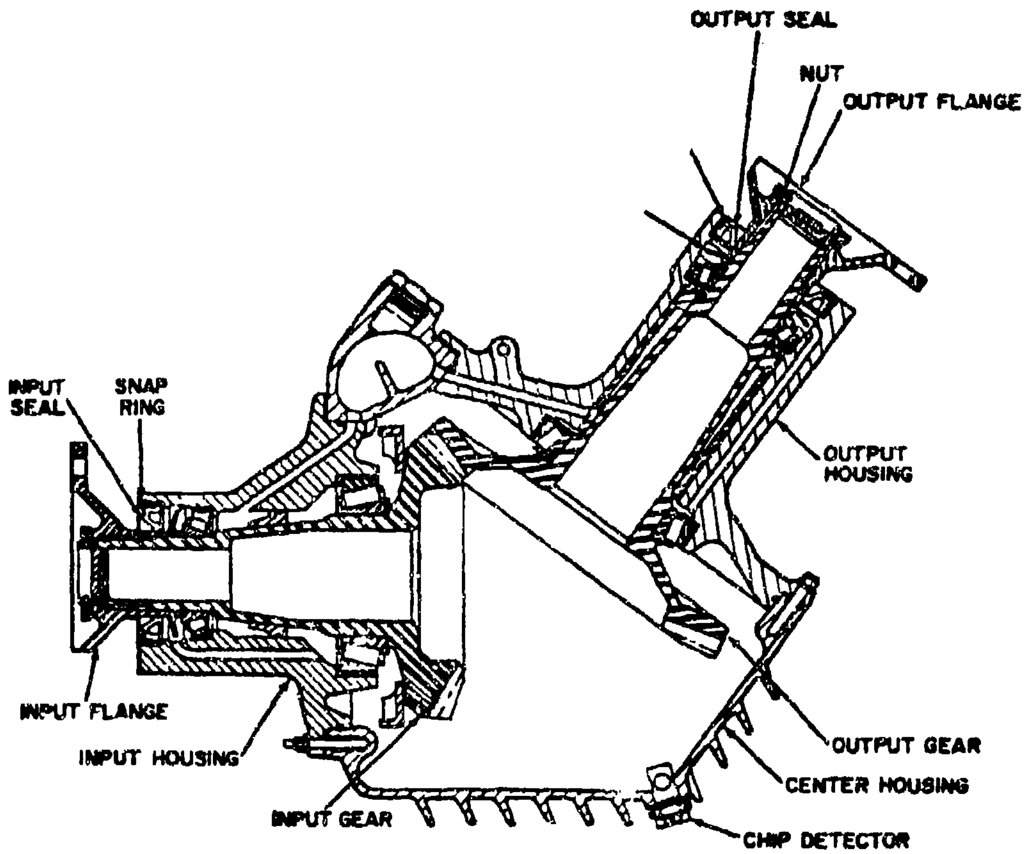

Figure 28. INTERMEDIATE GEARBOX WITH KNOWN MTBUR AND TBO

TABLE 3. RELIABILITY ANALYSIS FOR NEW GEARBOX

PART	i	QUANTITY (N)	FAILURE MODES	θ_i	β_i	$N(100/\theta_i)^{\beta_i}$	$N(10000/\theta_i)^{\beta_i}$	$R_i(100)$	$R_i(10000)$
Spiral Bevel Gears	1	2	Excess Wear	8.4×10^5	1.4	6.41×10^{-6}	4.05×10^{-3}	.999994	.995962
	2	2	Other Modes	1.3×10^6	1.0	1.54×10^{-4}	1.54×10^{-2}	.[illegible]846	.984733
Tapered Roller Bearings			Spalling						
	3	1	25,700 hours	1.9×10^5	10/9	2.27×10^{-4}	3.79×10^{-2}	.999773	.962765
	4	1	31,900 hours	2.4×10^5	10/9	1.75×10^{-4}	2.93×10^{-2}	.999825	.971154
	5	1	56,500 hours	4.3×10^5	10/9	9.18×10^{-5}	1.53×10^{-2}	.999908	.984805
	6	1	9,600 hours	7.3×10^4	10/9	6.58×10^{-4}	1.10×10^{-1}	.999342	.895979
	7	4	Other Modes	2.5×10^6	1.0	1.60×10^{-4}	1.60×10^{-2}	.999840	.984127
Lip Seals	8	2	Leakage	1.7×10^6	.5	1.53×10^{-2}	1.53×10^{-1}	.984778	.857793
O-Rings	9	5	Leakage	8.8×10^7	.74	2.00×10^{-4}	6.03×10^{-3}	.999800	.993992
Shaft	10	2	Crack/Fracture	2.0×10^7	1.0	1.00×10^{-5}	1.00×10^{-3}	.999990	.999000
Flange	11	2	Crack/Fracture	8.8×10^5	1.0	2.27×10^{-4}	2.27×10^{-2}	.999773	.977529
Housings	12	3	Crack/Fracture	9.8×10^4	1.7	2.47×10^{-5}	6.19×10^{-2}	.999975	.939930
	13	3	Corrosion	1.5×10^4	1.3	4.45×10^{-3}	1.77	.995561	.170174
				TOTAL (Σ)		.02172	2.2438		

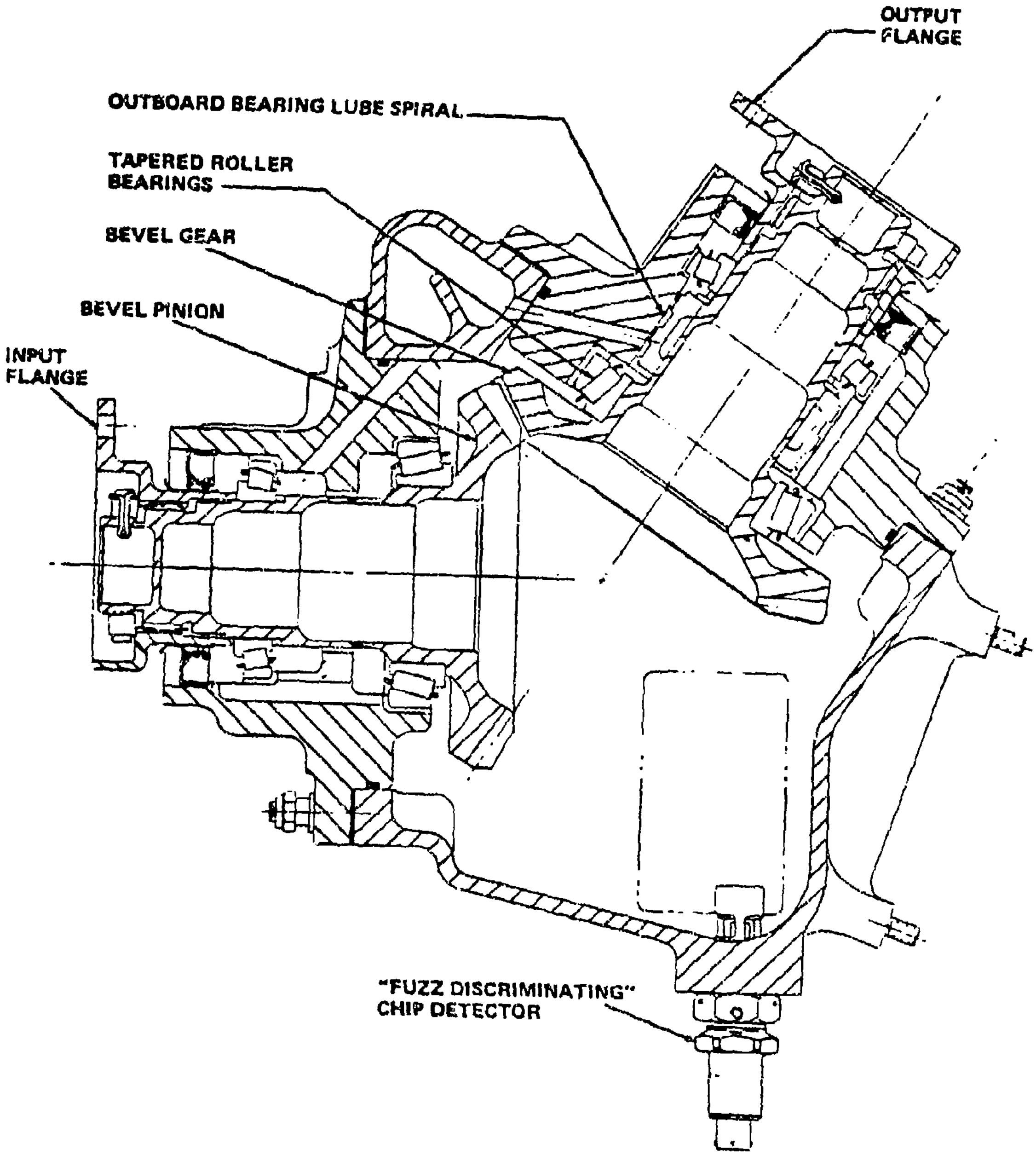

Figure 29. NEWLY DESIGNED INTERMEDIATE GEARBOX

STEP 4

Compute the theoretical MTBR for the existing gearbox using the shape and size parameters calculated in Step 4 by substituting in Equation (6).

$$\Gamma(1 + 1/1.003) = \Gamma(1.997) = 1.00 \qquad \text{(see Figure 27)}$$

$$MTBR = 4679(1.00) = 4679 \text{ hours}$$

STEP 5

Compute the $MTBUR_{adjusted}$ and the correlation factor for the existing gearbox using Equations (8) and (9). As previously stated, this gearbox has an observed MTBUR of 4000 hours and a TBO of 2000 hours.

$$MTBUR_{adjusted} = \left[4000(2000)^{1.003} - 1\right]^{1/1.003}$$

$$\times\ \Gamma(1 + 1/1.003)$$

$$MTBUR_{adjusted} = 3992(1.00) = 3992 \text{ hours}$$

$$K = (3992/4679)^{1.003} = .853$$

STEP 6

Compute the reliability of the new gearbox at 100 hours using Equation (10). Values of the exponent and the reliability without consideration of the correlation factor are given in Table 3 for each failure mode.

$$1/K \ \Sigma(100/\Theta_i)^{\beta i} = 1/.853(.02172) = .02547$$

$$R(100) = e^{-.02547} = .9749$$

STEP 7

Compute the reliability of the new gearbox at 10,000 hours using Equation (11). Values of the exponent and the reliability without consideration of the correlation factor are given in Table 3 for each failure mode.

$$1/K \ \Sigma(10{,}000/\Theta_i)^{\beta_i} = 1/.853(2.2438) = 2.6305$$

$$R(10{,}000) = e^{-2.6305} = .07204$$

STEP 8

Compute the composite shape and size parameters, β and Θ, for the new gearbox using the values calculated in Steps 6 and 7 by substitution in Equations (4) and (5).

$$\beta = \frac{\ln\ln\frac{1}{.07204} - \ln\ln\frac{1}{.9749}}{4.605} = 1.007$$

$$\Theta = e^{\frac{4.605(1.007) - \ln\ln\frac{1}{.9749}}{1.007}} = 3834$$

STEP 9

Compute the MTBR of the new design using Equation (6).

$$\Gamma(1 + 1/1.007) = \Gamma(1.993) = 1.00 \qquad \text{(see Figure 27)}$$

$$\text{MTBR} = 3834(1.00) = 3834 \text{ hours}$$

The correlation factor is also used to determine the probability of passing a test. This probability, P(T), can be determined using the following equation:

$$P(T) = P(\text{no failures in T hours}) = e^{-1/K\ \Sigma(T/\Theta_i)^{\beta_i}} \qquad (12)$$

Example - Probability of Passing a 200-Hour Test

If the new gearbox were subjected to a severe 200-hour test that reduced the L_{10} lives of the four bearings to 2142 hours, 2658 hours, 4708 hours, and 800 hours, the probability of passing the test can be determined using Equation (12). Setting T = 200, using the size and shape parameters of Table 1, and a correlation factor of K = .853, we have from Table 4:

$$1/K\ \Sigma(200/\Theta_i)^{\beta_i} = 1/.853(.07340) = .08605$$

$$P(200) = e^{-.08605} = .918$$

Hence, there is about a 92 percent chance of passing the 200-hour test.

TABLE [illegible]. RELIABILITY ANALYSIS FOR 200-HOUR TEST

PART	i	QUANTITY	FAILURE MODES	θ_i	[illegible]	$N(200/\theta_i)^{\beta_i}$	$R_i(200)$
Spiral Bevel Gears	1	2	Excess Wear	8.4×10^5	1.4	1.69×10^{-5}	[illegible]
	2	2	Other Modes	1.3×10^6	1.0	3.08×10^{-6}	.999692
Tapered Roller Bearings			Spalling				
	3	1	2142 hours	1.6×10^4	10/9	7.68×10^{-3}	.992348
	4	1	2658 hours	2.0×10^4	10/9	5.99×10^{-3}	.994023
	5	1	4708 hours	3.6×10^4	10/9	3.12×10^{-3}	.996885
	6	1	800 hours	6.1×10^3	10/9	2.24×10^{-2}	.977822
	7	4	Other Modes	2.5×10^6	1.0	3.20×10^{-4}	.999680
Lip Seals	8	2	Leakage	1.7×10^6	.5	2.17×10^{-2}	.978541
O-Rings	9	5	Leakage	8.8×10^7	.74	3.33×10^{-4}	.999667
Shaft	10	2	Crack/Fracture	2.0×10^7	1.0	2.00×10^{-5}	.999980
Flange	11	2	Crack/Fracture	8.8×10^5	1.0	4.55×10^{-4}	.999546
Housings	12	3	Crack/Fracture	9.8×10^4	1.7	8.01×10^{-5}	.999920
	13	3	Corrosion	1.5×10^4	1.3	1.10×10^{-2}	.989107
				TOTAL (Σ)		.07340	

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217.01 Information Sheet - Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears

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APPENDIX A

DESIGN CHECKLISTS

The checklists presented on the following pages are intended as a guide. The use of fairly detailed checklists during a design program can prevent the occurrence of costly oversights and insure a uniform approach in the design of helicopter gearbox components. The checklists here may be used as is, or new checklists may be made up according to the desires of the design manager. It is, however, strongly suggested that they be prepared before the start of a design program, and made an integral part of the design procedure.

<u>SPUR/HELICAL GEAR MESH DESIGN CHECKLIST</u>

	Pinion	Gear	Allowable
Nomenclature	______	______	
Part Number	______	______	
HP	______	______	
RPM	______	______	
Bending Stress	______	______	______
Hertz Stress	______	______	______
Flash Temp. Index	______	______	______
EHD Film Thick	______	______	______

( ) Balance design with respect to bending and hertz stress

( ) Design for recess action

( ) Adequate root fillet radius on both gear and pinion (.030 min)

( ) Correct case depth and hardness for pitch

( ) Adequate gear backup (min 1.15 x whole depth)

( ) Profile modification correctly specified

( ) Web centered under load and checked for adequate strength in event of uneven load distribution across face

( ) Gear diameters held concentric to bearing journal diameters

( ) Crowning specified for gears with high L/D ratios

( ) Mesh designed with hunting teeth

( ) Measuring pin diameter large enough to protrude above outside diameter but not contacting teeth in tip relief area

( ) Tip/root clearance adequate

( ) Backlash correct for pitch

( ) TIF diameter greater than base circle diameter, i.e., no undercutting

SPIRAL BEVEL GEAR MESH DESIGN CHECKLIST

	Pinion	Gear	Allowable
Nomenclature	____________	____________	
Part Number	____________	____________	
HP	____________	____________	
RPM	____________	____________	
Bending Stress	____________	____________	______________
Hertz Stress	____________	____________	______________

( ) Hand of spiral chosen so that axial force tends to push both pinion and gear out of mesh, or if not possible, so that pinion is pushed out of mesh.

( ) Face width limited to not more than 1/3 outer cone distance.

( ) Face contact ratio at least 1.25

( ) Correct case depth and hardness for pitch

( ) Gear diameters held concentric with bearing journals

( ) Adequate root fillet radius on both pinion and gear

( ) Crown to section dimension specified so gear blank can be properly checked

( ) Gear backup adequate (1.15 x whole depth)

( ) Webs in line with theoretical directions of resultant loads

( ) Acceptable tolerance for V and H check readings specified

BEARING DESIGN CHECKLIST

Location	______	Part Number	______
Type	______	Number per Aircraft	______
Bore	______	O.D.	______
Class	______	Bearing Material	______
Cage Material	______	Cage Type	______

	Vendor	Vendor Part No.	Capacity	L_{10}
1.	______	______	______	______
2.	______	______	______	______
3.	______	______	______	______

Lubrication Factor	______
Misalignment Factor	______
Material Factor	______
Adjusted L_{10}	______

( ) Consult bearing manufacturer on application

( ) Inspection requirements (magnetic particle, nital etch, etc.) specified

( ) Heat treatment and carburization requirements specified

( ) Max breakout adequate to clear shaft shoulder radius

( ) Shaft/bearing fit adequate to prevent creep at max load

( ) Crown of roller bearing consistent with max continuous load

( ) Internal clearance acceptable at all temperature extremes (ball and roller bearings)

( ) Duplex bearings clearly marked for correct installation

( ) Preload or end play correct (tapered roller bearings)

SEAL DESIGN CHECKLIST

Location ______	Part Number ______
Type ______	Speed ______
Shaft Eccentricity (Static) ______	Shaft Runout (Dynamic) ______
Housing Bore Eccentricity ______	Shaft Material ______
Shaft Hardness ______	Shaft Surface Finish ______
Temperature (Normal Operating) ______	Temperature Range ______
Pressure ______	Leakage Rate ______
Type Fluid Being Sealed ______	Seal Material ______
External Environment ______	

( ) Seal speed capability consistent with operating speed considering pressure and runout

( ) Seal pressure capability consistent with operating pressure considering speed

( ) Runout not excessive in application

( ) Extremes of seal axial position relative to shaft checked to insure adequate running surface

( ) Seal material compatible with fluid being sealed

( ) Primary seal protected from dust, grit, etc., by secondary seal

( ) Sacrificial runners considered for use on shaft

MANAGEMENT R&M CHECKLIST

( ) Reliability and maintainability goals clearly established

( ) Preliminary reliability analysis scheduled early in design stage

( ) Design review scheduled after completion of basic layout before detail design

( ) Detailed reliability analysis scheduled during detail design

( ) Consideration of "must pass" tests taken in design, i.e., must have acceptably high chance of passing test

( ) Attention to detail stressed in analytical effort

( ) High risk areas identified and alternate designs prepared

( ) Communication maintained with other design areas to minimize accessibility problems

( ) Allowance made for increased power (especially in tail rotor drive system components) so that excessive redesign will not be necessary

( ) Perform failure mode and effects analysis